POWER

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The Roadside Pump

AFTER we had graduated from candles and whale-oil to coal oil, and kerosene had become the popular illuminant, the grocer used to draw it from a barrel into a tin measure. It got onto the floor and onto his clothes and hands and its penetrating odor was absorbed by or communicated to the edibles that he handled.

And then came Bowser with the self-measuring pump.

When the first "horseless carriage" sputtered down the street it passed stores where these self-measuring pumps had been in use for many years. They were quickly adapted to handle gasoline, and the red wooden cabinets to which Bowser gave the name of "filling stations," which has clung to them all these years, began to appear on the curb

Did you ever try to realize what a million means?

If a clock ticks once a second it will have to run twelve days to tick a million times.

A million paces would carry a man from New York to Buffalo.

S. F. Bowser & Company, Inc., completed, in June of this year, their millionth self-measuring pump.

Its development has been parallel with that of the automobile, the needs of which have given it its greatest application and opportunity.

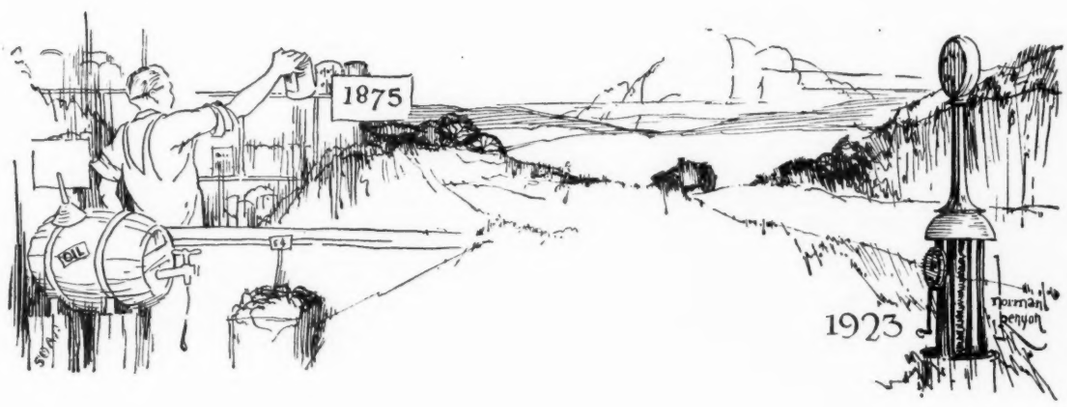
Others have taken up its manufacturing and now, all over the country, on the "big roads" and the back roads, these ingenious and handy devices, adapted to out-of-door service and quick, accurate and safe manipulation are gaudy welcome beacons to the driver who is getting out of gas.

The Bowsers shut up shop for a day and celebrated, which they were well entitled to do. I regret that I could not accept their kind invitation to participate.

I suppose they are well started on their second million by this time and it will not take as long to make the second as it did the first.

The first pump that they built has probably gone the way of all things, but the millionth ought to be saved to compare with the two-millionth when the hand has gone around again.

F. R. Low



Results of Seventy-Three Tests Made on Hydraulic-Turbine Governors*

THESE tests were conducted to determine how parallel operation of hydro-electric and steam plants forty miles apart could be improved. Among the elements investigated are insensitiveness of the governors, effects of kinetic energy in the machine's revolving parts and power-system reactance.

UNDER normal conditions, with the exception of certain periods on Sundays, hydro-electric units in the Holtwood plant of the Pennsylvania Water and Power Co., at Holtwood, Pa., operate in parallel with the steam units at the Westport and Pratt Street plants of the Consolidated Gas and Electric Co. at Baltimore, Md. The plants are connected for parallel operation through four 70,000-volt transmission lines.

There had been complaints from the main steam station that the Holtwood governors were too sluggish, which caused the steam units to handle all the system swings. It was claimed that sudden load increases on the system resulted in overloading of the steam units, while sudden decreases were likely to cause motoring. It was partly on this account that it was considered necessary to carry more load on the steam units in operation than the system load would normally justify. To reduce chances of motoring, the load on each 20,000-kw. steam unit was not allowed to drop below 5,000 kw. (See article "Recording Hydraulic-Turbine Governor Operation with an Oscillograph," in the Sept. 18 issue.)

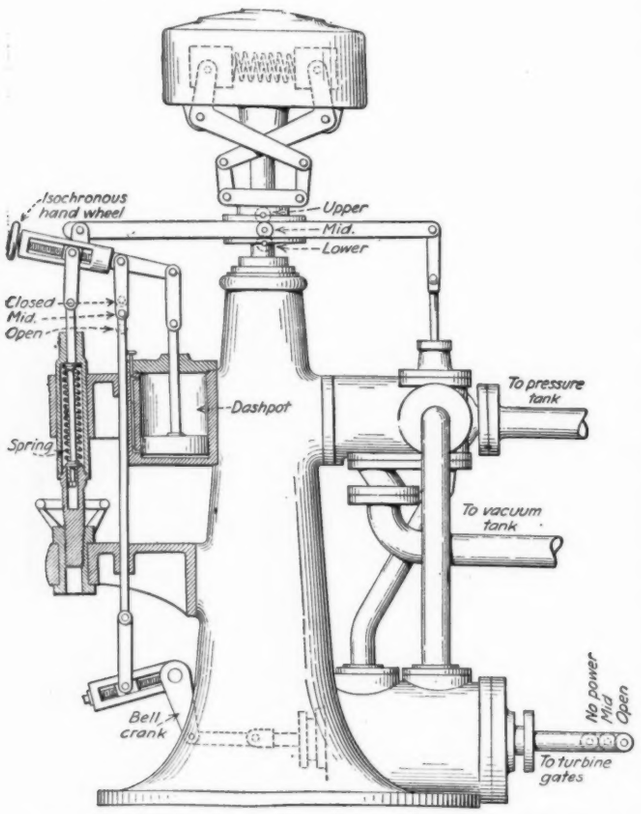


FIG. 1—TYPE OF HYDRAULIC-TURBINE GOVERNOR ON WHICH TESTS WERE MADE

This extra steam generation resulted in a considerable waste of water at the hydro-electric plant during high-flow periods, and it was in an attempt to improve these conditions that a series of tests on the Holtwood governors was started.

Failure of Hydraulic Units To Carry Their Share of the Load

The alleged failure of the Holtwood units to carry their share of the load swings it was believed might be due to either or all of the following causes:

A. Insensitiveness of the governors.

B. The smaller kinetic energy stored in the revolving parts of the individual units.

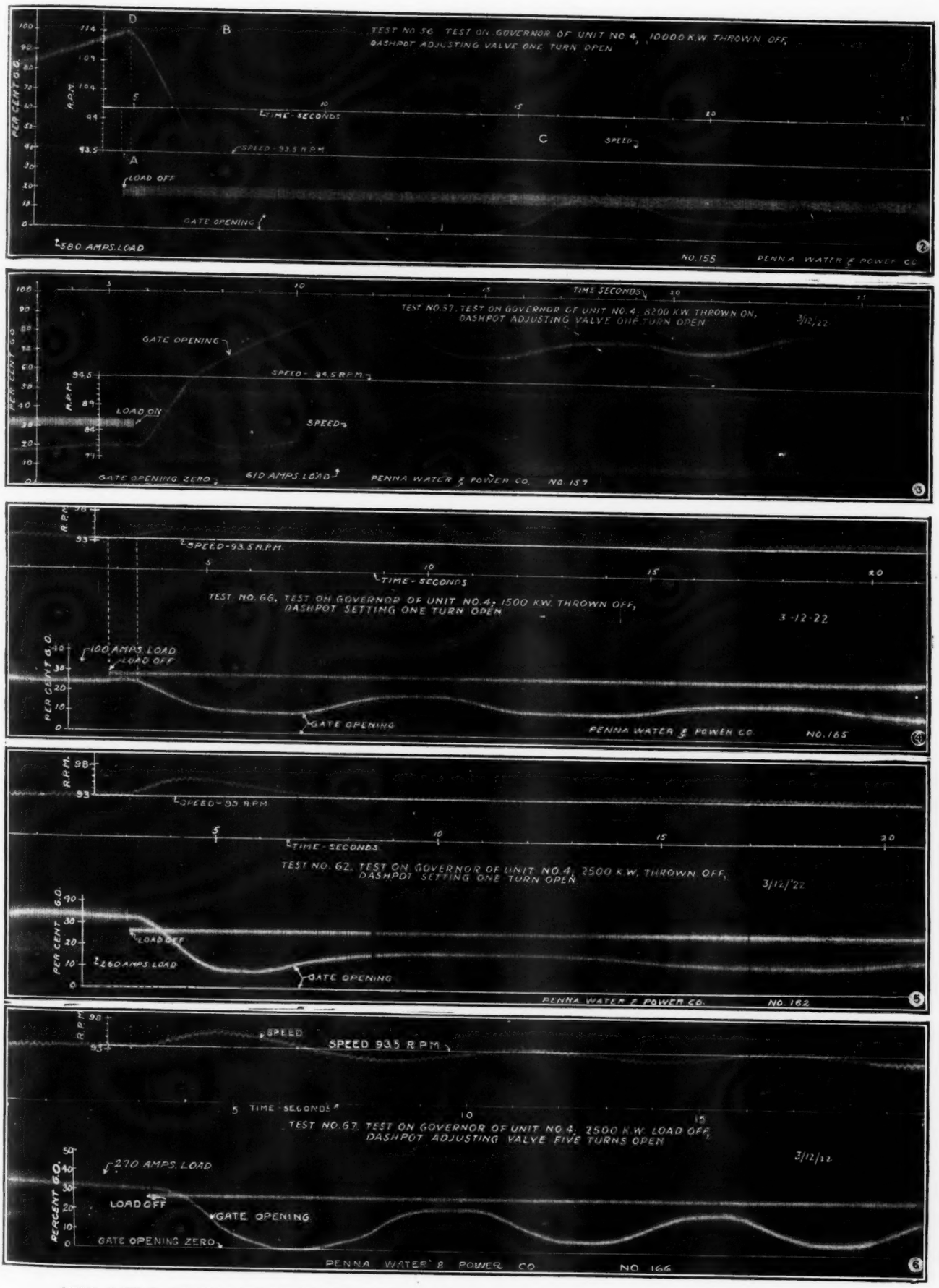
C. The possibility that the sudden swings might be considerably damped at Holtwood by the reactance of transformers and transmission lines through which this station feeds, compared with the more direct "feed" from Westport.

These various possibilities have all been investigated and will be considered in the order mentioned, though it is not possible to keep items A and B entirely apart, as the result of any governor action depends largely upon the kinetic energy stored in the system.

A. Insensitiveness—By insensitiveness is meant in general a measure of the governor's inability to maintain constant speed on the unit. It may be considered under two headings: (1) "Detention by friction," which prevents the governor from responding readily to load charges; and (2) "delay in restoration," which means the interval between the load change and the return to normal speed. The first of these is due to friction in the governor mechanism, while the second is in addition dependent on flywheel effect, fluctuation in wheel-pit pressure, traversing time of governor and relay[1] and the various governor adjustments. Both of these affect the distribution of load swings between the steam and hydraulic units. If the latter are slow in starting and restoring, the greater part of the large load swings will be shifted over on the steam units, and on small load swings the hydraulic governors may not get into action at all. Unequal division of load swings between steam and hydraulic units must to a certain degree be expected when they are operated in parallel, unless the frequency requirements of the system are such as to permit a slowing down in the action of the steam governors to make them correspond to the hydraulic governors.

*The data in this article were supplied by A. F. Bang, superintendent of operation, Pennsylvania Water & Power Company, under whose directions the tests were made.

[1]Relay in this article means the gate-operating cylinder (servomotor).



FIGS. 2 TO 6—OSCILLOGRAMS TAKEN OF CHANGES IN LOAD, SPEED AND GATE OPENING AS A MEANS OF STUDYING THE GOVERNOR'S MOVEMENTS

The action of the latter, especially with reaction-type turbines for high heads, cannot be speeded up beyond a certain point. On sudden load reductions by the unit it is unpractical to close the gates too quickly, because even if the mechanism will stand the strain, the sudden retardation of the large water column in the penstock will build up pressure in the wheel pit which may have an effect greater in proportion than the decrease in volume of water and actually result in momentarily speeding up instead of slowing down the unit, thus defeating improvement in regulation. Again, on sudden opening of the gates the water column requires a certain time to accelerate, depending on the slope of the penstock, and regulation is thereby limited in this direction.

Operation of Governors

The governors are water operated and the flyballs control the pilot valve which operates through one or more relay valves to move the gates in the proper direction, Fig. 1. The gates are in turn connected back to the pilot valve through links acting in connection with a spring and dashpot. Any movement of the gates tends to return the pilot valve to neutral, but the time required for them to do so is dependent on the dashpot adjustment. This adjustment, by controlling the time the pilot valve is permitted to remain open, fixes the new position of the gates which should exactly correspond to the new load.

As installed at Holtwood, the governors on units 1 to 5 are similar. Those on units 6, 7 and 8 differ from the others principally in their valve mechanism. These latter have one pilot valve and one relay valve, which are connected mechanically to a second pilot valve located below the governor base, which operates a second relay that in turn admits water to the operating cylinder. Unit No. 8 has two gate-operating cylinders instead of one, as used on all the other machines. In

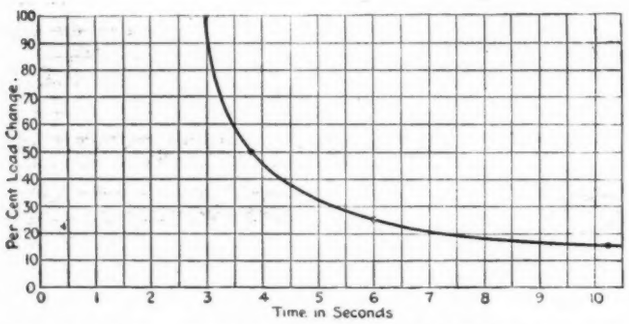


FIG. 7—CURVE SHOWING RELATION BETWEEN CHANGE IN LOAD AND GATE MOVEMENT

principle the governors are all alike and are adjusted in the same way.

There are three adjustments provided. The oil dashpot, which affects to some extent the duration of the sudden speed fluctuations; the isochronous handwheel used to regulate the static fluctuation; and the bell-crank adjustment, which is set at the time of installation to adapt the governor to the particular unit on which it is installed. This last adjustment is located in the series of links connecting the governor and the gate mechanism, and its action is to regulate the amount and duration of the pilot-valve opening during a load change, and accordingly affects the sudden fluctuations.

Points of interest to be noted in the design of these governors are the use of ball bearings in the link connections supporting the flyballs in contrast to the knife-edges commonly used in steam governors, and their heavy and sturdy construction throughout as compared to the latter.

Description of the Tests and Their Results

Altogether 73 tests, having a bearing on the insensitiveness of the governors, have been made, the more

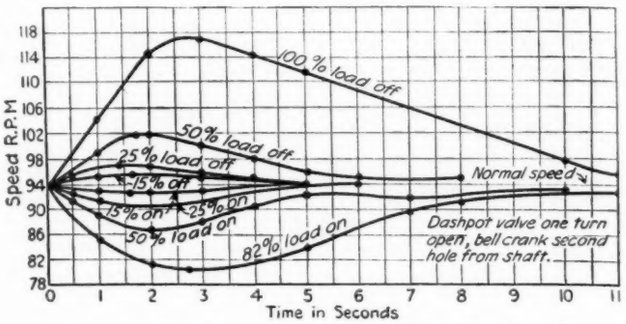


FIG. 8—CURVES SHOWING SPEED CHANGE FOR CORRESPONDING LOAD CHANGE

important of which will be considered in the following order:

1. General information on governor performance.
2. Decrease of detention by friction.
3. Increase in rate of gate travel.
4. Elimination of hunting.
5. Static fluctuation (regulation).

The oscillograph was selected as the best available means of securing a graphic record of governor performance. It was necessary of course to arrange that the various phenomena give an electrical indication. The exact method of doing this was described in detail in the previous article.

As the oscillograph is equipped with three elements only, the following three factors were selected for recording: (1) Speed of unit (r.p.m.); (2) position and movement of gates; (3) variation in wheel-pit pressure. The last-named factor was found to be quite small in this case and not of much importance; it was therefore in most of the tests replaced with a record of the (4) load on the generator in amperes.

Analysis of Oscillograms

Factors affecting the sudden speed fluctuations of a unit under load changes are:

1. Detention by friction of the governor.
2. Traversing time of governor and relay
3. Flywheel effect of the unit.
4. Variation of wheel-pit pressure.
5. Governor adjustments.

The oscillogram, Fig. 2, represents a 10,000-kw. load thrown off by the unit; Fig. 3 shows a sudden increase in load from 0 to 8,200 kw.; Fig. 4 a load rejection of 1,500 kw. to 0; and Figs. 5 and 6 give a comparison between two different settings of the dashpot adjusting valve of one turn and five turns respectively, on a load rejection of 2,500 kw. to 0.

In Fig. 2 the load is thrown off at point *A*, and the speed starts to rise immediately, until it reaches a maximum at point *B* just as the gates reach a position corresponding to no load. It is evident that up to this point excess power is being supplied to the unit, which

can result only in increasing the speed. The magnitude of this increase is a function of the flywheel effect of the unit, the time required after the load change for the gates to reach their new position and the wheel-pit pressure. The duration of the interval *A* to *B* is dependent only on the time required for the gates to reach their new position. The interval from *B* to *C*, or time required for the speed to be returned to near normal, is dependent on the flywheel effect of the unit, which now must be slowed down; and the governor adjustments, particularly the dashpot. Owing to the so-called static fluctuations of the governor, the speed does not return quite to its original value. This is done on purpose to obtain a better division of the load between the units when slow swings occur.

The gate does not start to move immediately as the load changes, but lags behind by an interval of time *A* to *D*. This time is partly due to detention by friction of the governor and partly to the necessary time required for the governor to reach a position where the pilot valve is open, as determined by the governor traversing time. It is evident that both these factors

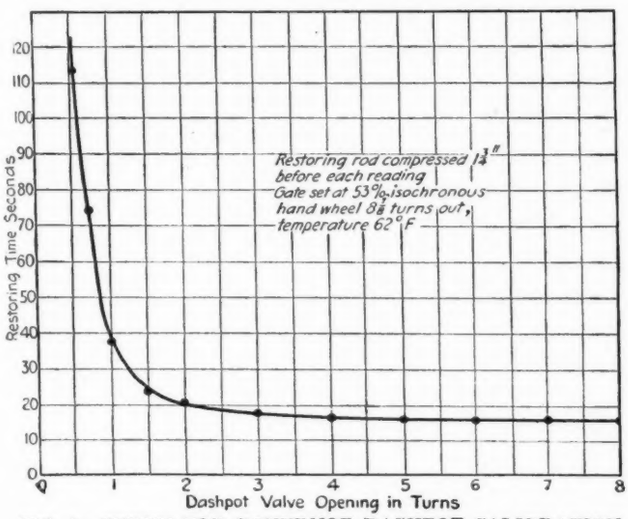

Restoring Time Seconds
Dashpot Valve Opening in Turns
Restoring rod compressed 1¾" before each reading
Gate set at 53% isochronous
hand wheel 8⅛ turns out,
temperature 62° F

FIG. 9—CURVE FOR GOVERNOR DASHPOT CALIBRATION

will cause the time interval *A* to *D* to vary with the magnitude of the load change, because of the different rates at which the speed fluctuates.

This can be clearly seen by a comparison of the oscillogram, Figs. 2 and 4, which represent load rejections of 10,000 and 1,500 kw. respectively. For the 10,000-kw. rejection the first movement of the gate occurs after about 0.2 sec., when the speed has risen about 1.5 r.p.m., against 0.6 sec. for the 1,500-kw. rejection with a corresponding speed increase of not more than 0.4 r.p.m.

It can be seen that the gate movement is a straight line for most of its travel, which means that during this time the governor pilot valve is wide open. The rounding off of the gate-movement curve at its upper and lower ends is due to the gradual opening and closing of the pilot valve. The more marked rounding at the lower end is traceable to the action of the governor restoring mechanism performing what is known in this governor as the "second compensation." Although the first and second compensations are supposed to be two distinct functions, the oscillograms show that they merge smoothly, and it is difficult to say at just what point the first is complete and the second starts.

From a study of the oscillograms it will be seen that the quickest rate of motion of the gates occurs when full load is thrown off, evidently corresponding to full opening of pilot and auxiliary valves. If the rate obtained at the beginning of the motion continued, full stroke would be covered in 2.7 sec.; that is, the relay traversing time is 2.7 sec. The traversing time obtained for smaller load changes is considerably larger (see Fig. 7 and the table). If less speed fluctuation is desired on sudden load changes, it is necessary either to cut down this time or to increase the flywheel effect.

Variation in wheel-pit pressure proved to be unimportant, because of the relatively low heads and velocities existing (see previous article). As far as wheel-pit pressure is concerned, it would be possible to increase the speed of the gate movement and thereby reduce the sudden fluctuation in speed on load changes. This has also been attempted in the newer units by increasing the diameter of the piston (units 6, 7 and 4) and the insertion of an extra relay valve giving larger opening to the gate-opening cylinder (units 6, 7 and 8) also in unit No. 8 by using two cylinders. It is probable that further improvements in this direction would be expensive and limited by the mechanical strength of the present pedestal bearings in the wheel pit.

Record of No. 4 Unit Performance

The table gives a condensed record of the performance of unit No. 4 as determined from oscillograms. From this it will be seen that the magnitude of the speed fluctuations varied all the way from 1.06 per cent on a 15 per cent load rejection, to 24.5 per cent for a 100 per cent load rejection, and the time elapsed before the speed was returned to near normal was 3.3 and 10.3 sec. respectively. Fig. 8, which is plotted from the oscillograms obtained, also gives a ready comparison between the speed, time curves obtained for various load changes from 15 per cent to 100 per cent both for load on and load off.

The effect of the dashpot adjustment is well shown by a comparison of the oscillograms, Figs. 5 and 6, representing a 2,500-kw. load change with dashpot adjustment of one turn and five turns open respectively. With the wider open dashpot, the speed returns to normal in 2.8 sec., while with the smaller opening 3.4 sec. is required. The former is an appreciable improvement, which will be greater on heavier load changes. Unfortunately, the wider open dashpot does not have the same tendency to prevent hunting as when slightly opened.

Damping Effect of Dashpot Adjustment

An idea of the damping effected by different adjustments of the dashpot is given on the curve, Fig. 9, which is plotted for turns of the adjusting screw, and the time required for the mechanism to return to its neutral position after being moved a measured distance away. This curve is for No. 4 unit, and data were secured by shutting the unit down, setting the gate at 53 per cent, and compressing and restoring rod spring by hand 1¾ in. When released, the spring acted to move the dashpot piston, and the time required for it to return to normal for the different adjustments was noted. It will be seen that little is gained by opening the adjusting screw more than three turns. It has been shown by experience, however, that it is unsafe to operate with a dashpot five turns open be-

cause of the severe hunting likely to follow a disturbance on the system.

It will also be seen from the table that the time elapsed between the load change and the first movement of the gate is somewhat erratic. This irregularity is probably due to the tendency of the governors to hunt, both while the unit is loaded and while idling. If the unit is hunting just before a test, the interval elapsing before the gates are first moved in the proper direction will be considerably affected by the direction in which the gates and governor mechanism are moving at the instant the load is changed. If this movement happens to be in the same direction as the gates will move to take care of the load change, the action is accelerated; if in the opposite direction, the action is retarded. An effort was always made to steady the governor down by hand before each test, but sometimes this was not successful. In general for 25 per cent load change and above this time element is on the average fairly constant, about 0.3 sec., evidently being dependent not so much on the "detention by friction" as on the necessary time required for the governor to open the valves, dependent on governor traversing time. For small load changes, below 25 per cent, this time element increases very fast; see oscillograms, Figs. 4 and 5, where time elements of 0.6 and 0.3 sec. are found for load rejections of 15 and 25 per cent respectively. In the light load changes the detention by friction evidently has a large influence in delaying governor action.

B. Comparison of Kinetic Energy Stored in the Hydraulic and Steam Units—As mentioned in the foregoing, the maximum variation in speed on a sudden load change depends essentially on the traversing time of the governor and relay (that is, gate cylinder) and the kinetic energy stored in the rotating masses. The kinetic energy depends in turn on the speed of the unit when the flywheel effect of the rotating masses is fixed. In the design of the hydraulic units, which are rated at 10,000 to 12,000 kw., the speed was fixed by other considerations, but great stress was laid on securing a large flywheel effect in order to obtain satisfactory governing. This was accomplished by using generators of large diameter with the poles fastened to a heavy steel or cast-iron ring acting as a flywheel. The flywheel effect obtained varies from 9,247,500 lb.-ft.[2] for units Nos. 6 and 7 to 18,227,500 for units 4 and 5. This is vastly more than the corresponding values for the 20,000-kw. steam units, which in the maximum case is only 256,000 lb.-ft.[3] Nevertheless, the much higher speed (about 16 times) at which the steam units run results in a 20,000-kw. steam unit having about 3.5 times the kinetic energy of the largest 10,000-kw. hydraulic unit (98,100,000 ft.-lb. against 27,400,000 ft.-lb.). In this respect a steam unit is far superior to a hydraulic unit, and in this fact undoubtedly lies a considerable part of the explanation of the better governing qualities inherent in the steam units. Even if all eight units at Holtwood are running and the benefit of their total kinetic energy is obtained, this is only 176,200,000 ft.-lb., or less than the kinetic energy contained in two 20,000-kw. steam units. Other advantages to the steam governors probably lie in a smaller governor and relay traversing time. The steam governors that use fly weights balanced on knife-edges probably also contain less friction.

C. Influence of Reactance—It had been suggested that since the steam plant is so much nearer the main load, there is much less reactance between the generators in this plant and the load than between the hydraulic plant and the load, and this might account for the steam units carrying the larger part of the momentary swings. An attempt was made at Holtwood to check up this theory, and tests indicated that when two units were properly balanced by means of the isochronous handwheel, they would divide the load equally both on steady load and on swings as obtained by throwing the load off and on. This load division was independent of whether they were in parallel directly or a large reactance inserted in one of the units as mentioned in the foregoing. It seems thus that if reactance causes any change in the load division, it is too quick to be observed on an indicating wattmeter.

DATA COMPILED FROM TESTS ON UNIT NO. 4

Oscillogram No.*	Load Change, Per Cent	Speed			Gate Movement				Gate Opening		
		Maximum Speed Change, Per Cent†	Time After Load Change That Maximum Speed Change Occurs, Seconds	Time From Load Change Until Speed Returns to 99 to 101 Per Cent Normal, Seconds	Time Between Load Change and First Movement of Gates, Seconds	Time for Gates To Move From Avg. Opening Before to Avg. Opening After Load Change, Seconds	Time for Gates To Reach Extreme Position After Load Change, Seconds	Relay (Gate Cylinder) Traversing Time, Seconds	Initial Value, Per Cent	Final Value, Per Cent	Maximum Change, Per Cent
149	26– 0	+3.7	1.4	4.5	0.25	1.4	2.4	5.9	37	17	27
150	0–25	—3.7	1.5	3.8	0.30	1.0	2.5	7.0	24	38	21
151	93–47	+11.7	2.0	3.9	0.20	1.8	2.9	3.1	100	40	83
152	49–93	—8.0	2.2	7.5	0.40	2.6	4.7	4.8	47	90	53
153	99– 0	+21.2	Off film	9.0	Off film	2.7	8.0‡	3.0	100	10	100
154	0–86	—18.6	3.0	9.5	0.30	4.2	7.5	3.5	17	80	83
156	0–58	—10.5	2.3	5.0	0.30	1.7	3.7	3.8	19	60	54
155	100– 0	+24.5	2.6	10.3	0.30	2.8	9.3§	2.8	100	14	100
157	0–82	—14.5	2.7	8.7	0.25	3.5	6.1	3.2	20	80	76
158	100–58	+13.3	2.3	4.9	0.55	1.6	3.3	2.8	100	52	84
159	57–86	—3.72	2.4	5.2	0.30	1.3	3.4	5.3	55	75	43
160	50– 0	+8.2	1.8	5.2	0.25	1.5	3.3	2.9	50	17	50
161	0–50	—7.75	2.2	4.9	0.25	1.5	3.3	4.2	19	50	47
162	25– 0	+3.2	1.3	3.4	0.30	1.2	2.5	6.5	33	16	25
163	0–25	—3.9	1.7	4.1	0.30	1.2	2.6	5.6	17	33	26
164	0–14	—1.7	1.9	4.6	0.60	1.3	3.2	10.0	17	27	14
165	15– 0	+1.06	1.7	3.3	0.60	1.1	3.5	10.5	28	17	18
166	25– 0	+3.7	1.5	2.8	0.30	1.3	3.1	5.4	33	18	33
167	0–26	—3.7	1.7	3.3	0.25	1.1	2.8	5.6	17	35	30

* Oscillograms 149 to 165 were taken with the governor bell crank connected in the second hole from shaft and dashpot one turn open. Head 52 ft. Oscillograms 166 and 167, taken with bell crank in second hole and dashpot five turns open.

† From tachometer readings.

‡ A 10 per cent gate opening was reached in 3 sec. and the gates closed in 8 seconds.

§ A 10 per cent gate opening was reached in 3.4 sec. and the gates closed in 9.3 seconds.

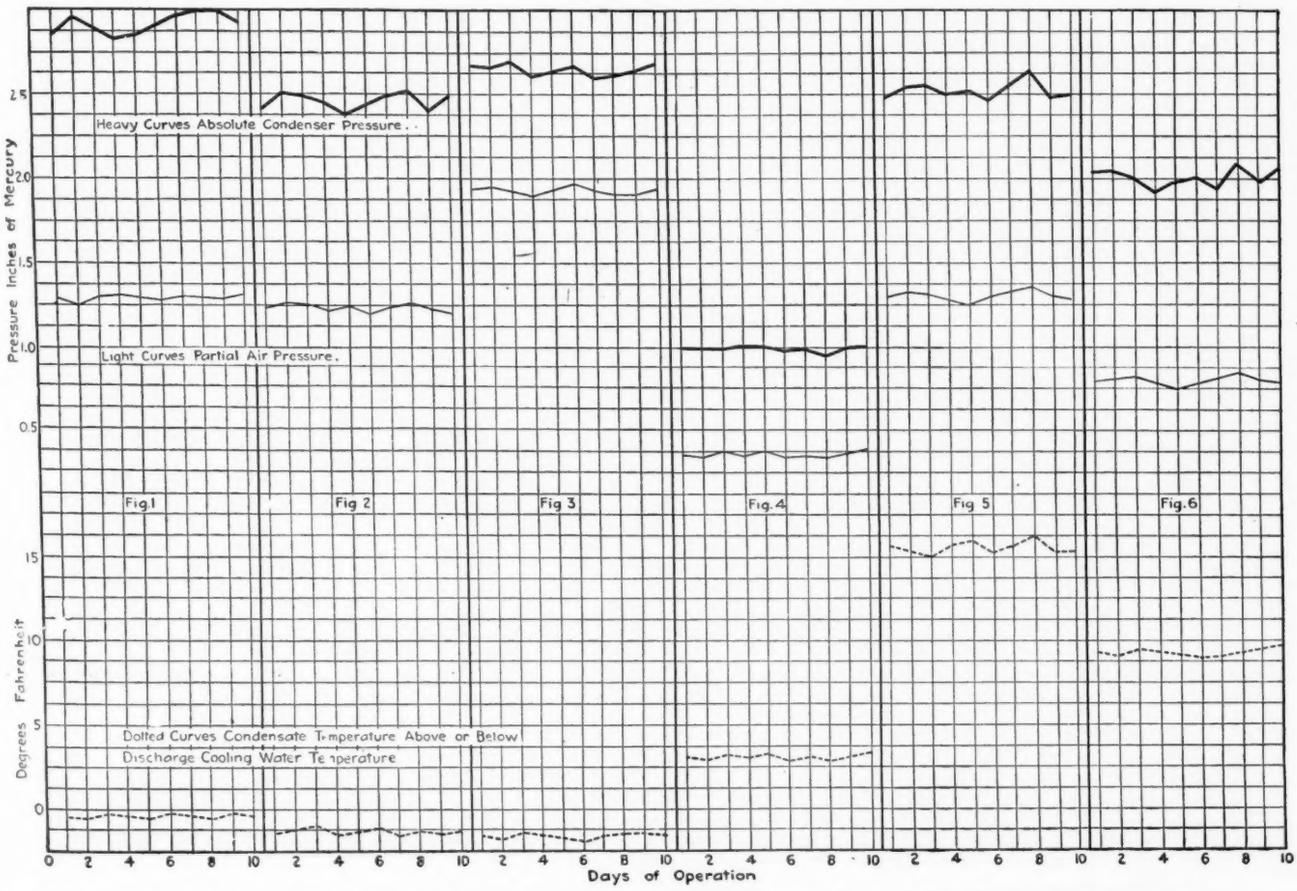
Condenser Records Show Air Leaks, Tube Scale or Good Performance

By S. THOMAS
Efficiency Engineer, Day & Zimmerman

In a previous article, "Determining Efficient Condenser Operation and Causes of Low Vacuum," which appeared in the Sept. 11 issue, Mr. Thomas described in detail a convenient method of calculating results and interpreting condenser records. This method is here shown as applied in practice to a 16,000-sq.ft. condenser serving an 8,000-kw. turbine. The top curves in the figures show if performance is good or bad, and the lower curves indicate the causes. Seven per cent fuel consumption was saved by eliminating an air leak indicated in Fig. 3.

IN ORDER to obtain the highest efficiency from condensers, a systematic record of their performance should be kept. The curves in Figs. 1 to 6 show the results obtained from a 16,000-sq.ft. surface condenser serving an 8,000-kw. turbine. Ten points, representing ten days' operation, are given on each set of curves, and each point is the average of five readings taken one-half hour apart, when the load on the turbine was practically constant at 8,000 kw. The heavy top curves show the absolute pressure in the condenser or the pressure against which the turbine must exhaust. The middle lines show the partial air pressure and the bottom dotted curves the amount by which the condensate temperature is above or below the discharge-water temperature. These indicate the effect of the three principal causes of low condenser efficiency; namely, an insufficient supply of circulating water, air leakage and dirty condenser tubes.

The first set of curves Fig. 1, show operation with an intake temperature of 73 deg. F. and a ratio of 45—that is, 45 lb. of circulating water per lb. of steam condensed. In Fig. 2 the improved results with the same intake temperature occurred after the second circulating pump was started, which gave a ratio of 74. The condenser pressure was reduced about ½ in., also the air pressure and condensate temperature were reduced slightly. The air pressure is still a great deal too high



FIGS. 1 TO 6—UPPER CURVES SHOW GOOD OR BAD OPERATION; LOWER CURVES INDICATE CAUSES

Fig. 1—Water ratio 45, intake water 73 deg. F., produce high condenser pressure. Fig. 2—Water ratio 74, intake water 73 deg., condenser pressure reduced 0.5 in., but some air leakage present with high partial air pressure. Fig. 3—Water ratio 67, intake water 55 deg., high partial air pressure shows air leak. Fig. 4—Water ratio 91, intake water 55 deg., normal partial air pressure and condenser pressure showing minimum air fairly clean tubes and standard performance generally. Fig. 5—Water ratio 93, intake water 75 deg. High condenser pressure and condensate rise showing dirty tubes. Fig. 6—Water ratio 95, intake water 75 deg., and lower condensate rise indicate cleaner tubes but not clean enough.

on account of an air leak. With this particular turbine operating at full load, an increase of one inch in vacuum between 28 and 29 in. causes a 4 per cent reduction in the steam consumption. The steam required to operate the second circulating pump is about 1.10 per cent of that consumed by the main unit. Then a gain of 0.275 in. in vacuum due to operating the second pump will just balance the cost of running it. Therefore, under the conditions cited, with a gain of ½ in. in the vacuum, it pays to run the second pump.

High Partial Air Pressure Shows Air Leak

The set of curves in Fig. 3 show a very bad air leak. The intake temperature was 55 and the ratio was 67. The leak was caused by a bad gasket in the equalizing pipe line between the turbine and condenser. Immediately after this leakage was stopped, the condenser performance appeared as in Fig. 4, with a ratio of 91 and the same intake temperature as before. The condenser pressure was decreased about 1.7 in., and the air pressure was decreased 1.6 in., but the condensate temperature was raised about 5 deg. This is good performance and there is but little room for improvement, except possibly the tubes are a little dirty, as the condenser had been in service about eight months without being thoroughly cleaned. With best performance, an air pressure of about 0.3 in. can be obtained, whereas the above air pressure was 0.36. The effect of eliminating the air leak on this condenser was very noticeable in the total station economy, decreasing the fuel consumption about 7 per cent.

In Fig. 4, if only one pump had been operating, the ratio would have been 52 and the absolute pressure 1.30 in. As a gain of 0.3 is obtained by running the second pump, a slight net gain in total economy is made and it probably pays to keep both pumps in service. However, it should be noted that the second pump does not cause as much gain in vacuum in this case as in the case of Fig. 2. This is true because the intake temperature is 18 deg. lower. The lower this temperature the less the gain obtained by pumping more water.

The performance of this condenser, Fig. 5, occurred about a year later, when the tubes were very dirty, with an intake temperature of 75 and a ratio of 93. There is a big increase in the condensate temperature, the air pressure and the condenser pressure.

The results shown in Fig. 6 were obtained after the condenser was cleaned, with a ratio of 95 and the same intake temperature as before. The performance is better than it was, but not so good as it should be. The condensate temperature, being so high, indicates that the tubes are still dirty and therefore were not cleaned very well.

Low Upper Curve Denotes Good Operation

It should be noted that the top curve shows what results are obtained, whether good or bad, while the other two curves tell what causes the results to be good or bad. In the first three sets of curves, where there is considerable air leakage into the condenser, the air pressure is high and the condensate temperature low. This is logical because the greater the amount of air present the lower is the temperature in the steam space. The curves Fig. 4 are taken as the standard of performance for this condenser, as they show the best results obtained in daily operation, although they could probably be improved slightly. In other condensers the two top curves come about the same as these, but the bottom curve varies with the tube spacing, steam lanes, etc., in different designs. Figs. 5 and 6 show a high air pressure, but the fact that the condensate temperature is so high indicates that this is due more to dirty tubes than to air leakage.

The total absolute pressure in the condenser may be considered as being made up of two parts—the vapor pressure at the temperature of the discharge water and the partial air pressure. Theoretically, the vapor pressure at the temperature of the vapor inside the condenser should be used, but as it is extremely difficult to determine such temperature, the discharge temperature is taken as next best. As explained before, the partial air pressure can be reduced by stopping air leaks or by cleaning the condenser tubes. It should always be kept as near 0.3 in. as possible.

Vapor Pressure Rises with Higher Intake Temperature

The vapor pressure can be reduced by pumping more water or by using water of a lower temperature. We have seen, by comparing Fig. 2 with Fig. 1, how the vapor pressure or the space between the top curve and the middle curve below was reduced by pumping more water. Now, by comparing Figs. 4 and 5, you will notice that the vapor pressure is considerably less in the former, although about the same amount of circulating water is used in both cases. This is due to the fact that the intake temperature, Fig. 4, was 55 deg., while it was 75, or 20 deg. higher in Fig. 5. As the temperature of the intake increases, the vapor pressure rises rapidly, and during the summer months it becomes so great as to very materially reduce the vacuum. About the only practical thing that can be done to reduce it is to pump more water, and of course the economical limit in this direction is soon reached.

Such a set of curves for each condenser, presenting a sort of picture of their performance, is a valuable record. It gives a good clue as to what is wrong when the condenser is not operating properly and indicates about how much can be gained by correcting the bad features. The writer has found them to be a great help and they should be of considerable interest and assistance to anyone trying to obtain the highest efficiency out of the equipment he operates.

Equal distribution of load current among all the brushes on a motor or generator cannot be too greatly emphasized and constitutes one of the most important reasons for the use of flexible connections between the brushes and holders. Unequal distribution of current not only causes serious injury to the brushes which are carrying an excessive share, but through the glowing and sparking so produced may cause serious injury to the commutator, necessitating expensive repairs. Even where shunts are used, unequal distribution will sometimes exist if they are not properly connected to the brush or if the connection to the brush-holder is poor, whether due to dirty contact or to a loose screw. Troubles due to unequal distribution on machines where the brushes are equipped with shunts may often be eliminated by cleaning and tightening the contacts of all the pigtail terminals.

The average coal consumption of the British central power stations for last year was 2.78 lb. per kilowatt-hour.

Expansion in the Steam-Engine Cylinder

By MARSHALL L. BARKER*

IT IS a generally accepted theory that expansion in a steam engine is by no means adiabatic. By reason of the re-evaporation of the water resulting from the initial condensation of part of the steam feed, the expansion curve of an indicator diagram is assumed to give at any volume a higher pressure than expected with adiabatic expansion and causes the curve to approximate closely the rectangular hyperbole. The re-evaporation is the result of heat absorption from the cylinder walls, and the amount of initial condensation is dependent on

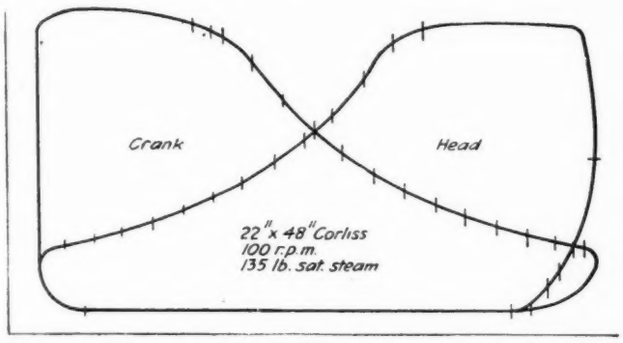


FIG. 1—DIAGRAM OF ENGINE WITH TIGHT VALVES

the cylinder-wall temperature during admission and, so it has been claimed, it is this condensation and re-evaporation loss which reduces the efficiency of an engine 15 to 50 per cent below that possible with a perfect or Rankine engine.

It is not and hardly can be definitely known just when this re-evaporation takes place, but as a result of the study of some 110 indicator diagrams covering nine makes of unaflow and ten Corliss engines, the author is inclined to doubt the accuracy of these well-accepted theories and to believe that the time from cutoff to release is too short for the water to absorb the heat from the walls. It is his opinion that the re-evaporation will take place after release, when there is more time, and a greater pressure and temperature drop from the original conditions. It is the latter supposition which explains more definitely the reason why adiabatic expansion can be obtained in a steam engine when the valves are steam tight.

These engine indicator diagrams plotted on logarithmic paper, along with what was known of the valve conditions and steam economy, lead one to believe that adiabatic expansion, or at least an expansion line that agrees with the adiabatic for the prevailing steam conditions, can be and is obtained in any steam-engine cylinder where the valves, piston, etc., are tight. This applies from half-load up, and in some cases even as low as friction load, but sufficient data are not available to prove conclusively that this always holds true for friction loads or for even light loads.

Several years ago J. Paul Clayton introduced this method of plotting diagrams on logarithmic paper. The method and reasoning employed are as follows: Any curve that has the general form $X\ Y^n = \text{constant}$ may be represented on logarithmic paper as a straight line making an angle with the horizontal, the tangent of which is equal to n. Expansion and compression of any elastic fluid, as steam for instance, follows this law as expressed in the form $P\ V^n = C$, where P is the absolute pressure in pounds per square inch, V is the volume in cubic feet, and a value of n depending upon the particular gas. Inasmuch as steam follows this law within narrow limits, the expansion and compression line of indicator diagrams can be plotted on logarithmic paper and any variations easily noted.

In general, leakage into the cylinder during expansion will raise the toe of the diagram, or the terminal pressure, and will lower the value of the exponent n, while leakage out of the cylinder will have the opposite effect. On the compression stroke leakage into the cylinder will raise the final compression pressure and also increase the exponent, while leakage out will do the reverse. The same amount of leakage will be much more noticeable on

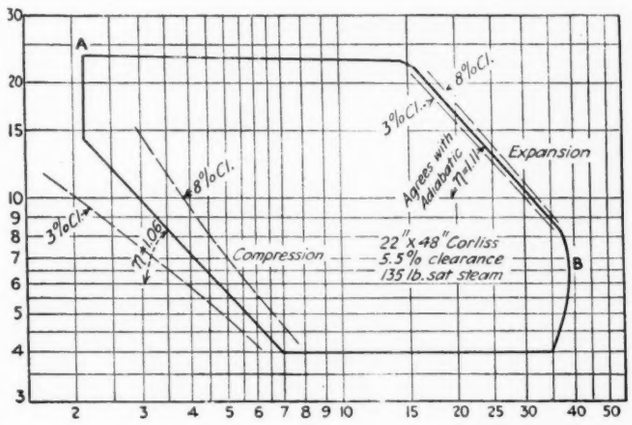


FIG. 2—LOGARITHMIC CHART OF FIG. 1

the compression stroke than on the expansion stroke, because the weight of steam is smaller and the leakage forms a much larger percentage of the weight of the steam present in the cylinder. Likewise, a light-load diagram will show leakage much more plainly than will a full-load diagram.

One of the preliminary steps in the investigation was to plot lines showing the adiabatic expansion of the steam from different initial pressures, with various degrees of superheat and quality. These lines served as references for diagrams having similar pressures and qualities of steam at admission. Ellenwood's steam tables and charts were used since

THE author has examined indicator diagrams from a large number of engines of all classes. As a result he arrives at a conclusion that if true throws a long cherished idea into the discard. "Power" will be interested in views of other engineers. In a second article the author will show a number of other diagrams from engines of various types.

*Engineer, Hooven, Owens Rentschler Co.

these include steam volumes. The points were obtained by following a constant entropy line passing through the original point and finding the volumes corresponding to pressures at different points. These volumes were then used to plot a logarithmic curve. Dry saturated steam gave practically the same exponent as 95 per cent dry. With steam initially wet, say 5 per cent moisture, or with dry saturated steam the exponent varies from 1.105 at 125 lb. to 1.10 at 200 lb. and the line is practically straight along its entire course. For 100 deg. superheat the line curves somewhat, but the average gives an exponent of 1.149. For 200 deg. superheat the line has a marked curvature, but the average is about 1.21, so for practical working conditions the value may be called 1.10 for saturated steam and 1.14 for 100 deg. superheat.

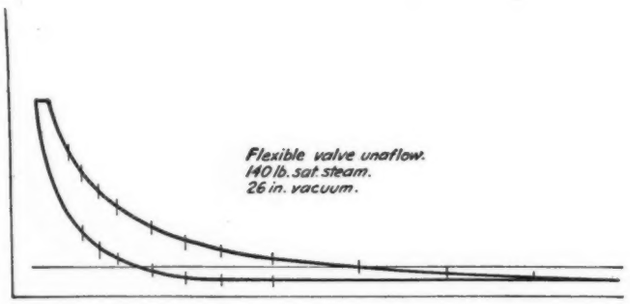


FIG. 3—DIAGRAM FROM FLEXIBLE-VALVE ENGINE

The limits of error for the exponents can be assumed very safely as plus or minus 0.02 for the expansion line and plus or minus 0.04 for compression, if it is of sufficient length and the clearance 3 per cent or greater.

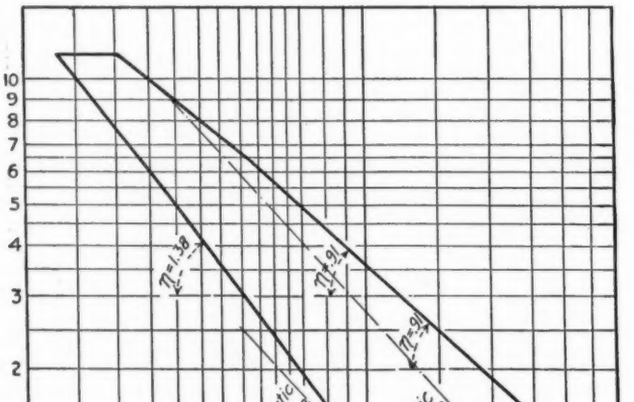


FIG. 4—DIAGRAM SHOWS VALVES LEAK

The exponent is easily obtained from the logarithm chart by laying off 1.0 inch to the left of the line horizontally and the vertical distance up to the line again will be the exponent, which is simply the tangent of the angle the line makes with the horizontal or the pressure line.

The indicator diagram, Fig. 1, from a 22x48-in. simple Corliss, is a good example of an engine with tight valves. As shown in Fig. 2, it gives an exponent for expansion that agrees with the adiabatic and a satisfactory compression line. The dotted lines were put in to show how the diagram would appear if the wrong clearance were used, and indicates that an error in clearance would show mainly in the compression line and would not seriously alter the expansion line. It will be noticed that the compression and expansion are practically parallel and that n is approximately 1.10.

Fig. 4, which is a transformation of the diagram in Fig. 3 shows steam-valve leakage quite plainly and is a

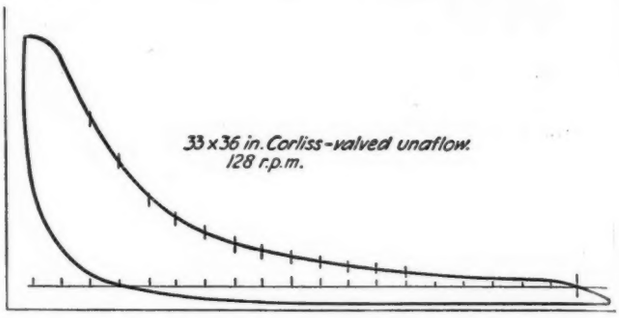


FIG. 5—CORLISS-VALVED UNAFLOW DIAGRAM

characteristic diagram for such conditions. It will be noted that the value of n for expansion is very low, being 0.91, while the n for compression is high, being 1.38, both indicating steam leaking into the cylinder. The lines are not parallel and diverge widely at the bottom.

A diagram, Fig. 5, from a 33x36-in. Corliss-valved uniflow with unjacketed heads, shows a very common condition when drawn on the logarithmic paper. The first half of the expansion line, Fig. 6, and the compression line show the valves to be tight and agree with the adiabatic, but the lower part of the expansion line shows steam-valve leakage, according to the writer's ideas. It is often interpreted as re-evaporation and is so common as to be very confusing in that respect. However, it must be remembered that at the end of expansion the piston is moving more slowly, giving more time for leakage, and the pressure difference is greatest across the valve, so if leakage was to occur it would be at this point.

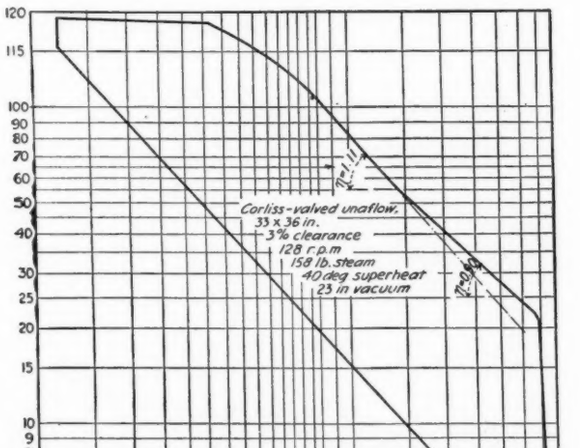


FIG. 6—VALVES LEAK TOWARD END OF EXPANSION

Another point to be noticed on this diagram is the absence of leakage indicated by the compression line, although evident in the expansion line. This is often even more pronounced and can be accounted for only on the supposition that valve leakage is stopped during the compression period.

It may be noticed that the compression does not appear to begin at 10 per cent of the stroke, but nearer 30

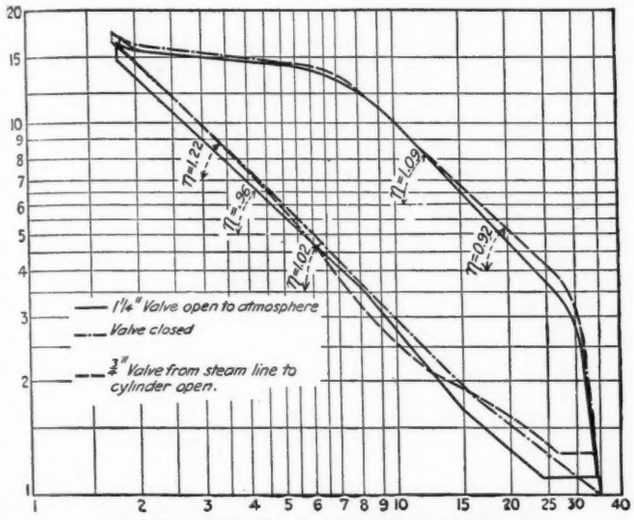


FIG. 7—EFFECT OF CYLINDER LEAKAGE

per cent. This is often found on condensing-engine diagrams where the heavier spring scale required does not react to the small difference in pressure at lower vacuums. The pencil sticks in a position part way down to vacuum, say 2 from 23 in., and does not start to rise on the compression line until the pressure reaches 21 in. again.

It is very seldom that an indicator with an 80- or 100-lb. spring will show the correct vacuum on the diagram within an inch or two of mercury, always being above the correct vacuum in the cylinder. The log diagram magnifies the lower part of the indicator diagram, otherwise the one or two inches would not be noticed.

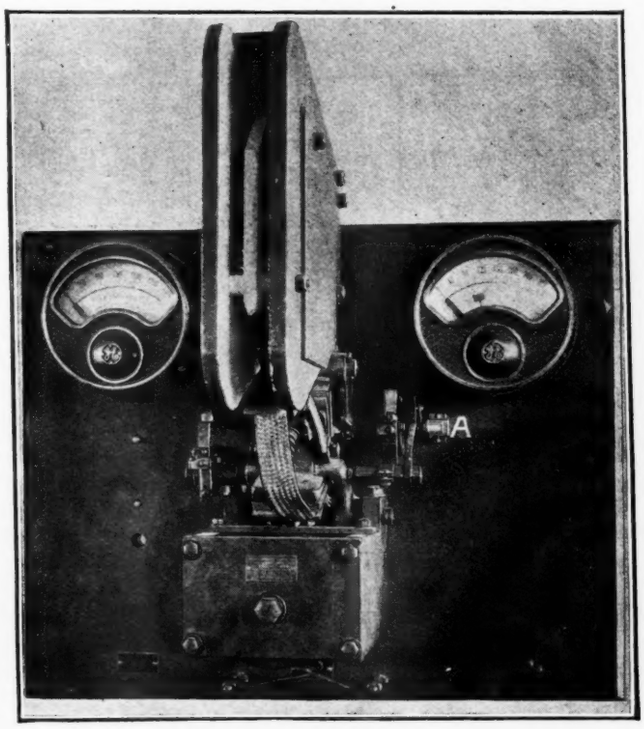
A point of interest that might be brought up is the amount of leakage required to cause a certain change in the value of n. A 26-in. unaflow cylinder was run at three-quarters load and diagrams taken with steam blowing into the cylinder from the steam line through a ¾-in. standard pipe connected to the relief-valve top. Then it was run with normal operation and finally with a 1¼-in. valve wide open from the cylinder to the atmosphere. These valves were arranged so they could be opened and closed quickly and the diagrams taken in rapid succession. The value of n was only changed from 0.92 in the first case to 0.96 in the second and 1.09 in the third for expansion. The area through a ¾-in. pipe is 0.53 sq.in. and through a 1¼-in. pipe it is 1.50 sq.in. of area, which would be equivalent to lifts of about 0.008 in. and 0.025 in., respectively, for a 9-in. double-beat poppet valve. So the amount of leakage to give an exponent as low as 0.80 can be imagined. The steam valve was already leaking some in this engine but that does not affect the relative results.

The heavy unbroken lines in Fig. 7 show that even with excess leakage the expansion line may be practically straight on the log paper; so the value of the exponent is the real criterion. The compression line will always be found curved as is shown by the unbroken line, denoting exhaust-valve leakage also. The steam-valve leakage and exhaust-valve leakage may produce a straight line if the rate of leakage of the two is in the right proportion.

The writer does not believe steam consumption can be determined from any indicator diagram, especially if the estimator does not know from what type of engine it was taken or has had no previous test experience with that type of engine. However, if an engine is once tested and the plotted diagram found correct, then future diagrams will show definitely whether the engine's efficiency is improving or decreasing, by the exponents and the appearance of the log lines, because the valves and piston are the chief factors in an engine's maintained economy. If it develops too much friction, it is soon evidenced by an overheated bearing, but valve leakage can go on undetected. One good method of detecting valve leakage is by the size of the loop on a non-condensing friction diagram on other than two-valve unaflow engines.

Counter Used To Record Opening of Circuit Breakers

The number of operations performed by the various opening and closing devices on automatic switching equipment is being recorded in an interesting manner by means of a standard feeder counter connected to such devices on installations that have been made



METHOD OF MOUNTING COUNTER ON CIRCUIT BREAKER

recently by the General Electric Co. The figure shows one of these counters at *A*, connected to the circuit-breaker mechanism of an automatic reclosing equipment. It will give an accurate record of the number of times this breaker opens or closes. By installing counters in connection with relay, contactors, etc., a complete check of operation may be shown which will lead to locating any faults.

Operation of Static Transformers*

How Voltage Transformation Takes Place—Determining the Value of the Primary and Secondary Voltages and Currents—Rating of Transformers—Connecting the Windings for Different Voltages

BY B. A. BRIGGS

ONE of the advantages of alternating current is the ease with which its voltage may be changed by the use of static transformers. The equipment has no moving part, consisting in its simplest form of a laminated iron core with two insulated windings on it and the whole immersed in a tank of oil. If these windings have a different number of turns, by applying a given voltage to one winding a voltage of another value will be obtained from the other winding. The winding to which voltage is applied is called the primary, irrespective of whether this is the high-voltage or the low-voltage winding. Conversely, the winding in which voltage is produced is called the secondary, whether this be higher or lower than the applied voltage. When a lower voltage is obtained at the secondary than is applied to the primary, the transformer is said to be a step-down transformer, which is the way a very large percentage of all transformers are used, except on high-voltage transmission lines. When the secondary voltage is higher than the primary, then the transformer is termed a step-up transformer. In general, for small and medium-sized transformers the primary is the high-voltage winding and consists of a larger number of turns of relatively small wire. The secondary consists of a small number of turns of large-sized copper to carry the relatively large secondary current.

A simplified diagram of a static transformer is shown in Fig. 1, and it will be assumed that it is in operation to step the voltage down. For simplicity assume that the primary winding has 2,300 turns and that this winding is connected to a 2,300-volt circuit. This voltage will cause a current to flow in the winding and magnetize the core. As this is an alternating current, the core will be magnetized and demagnetized with each reversal of the current. As the magnetic field builds up and decreases, it is being cut by the primary turns, which causes a voltage to be generated in it opposite to the applied voltage. The voltage induced in the primary winding of a well-designed transformer is almost equal to the applied voltage,

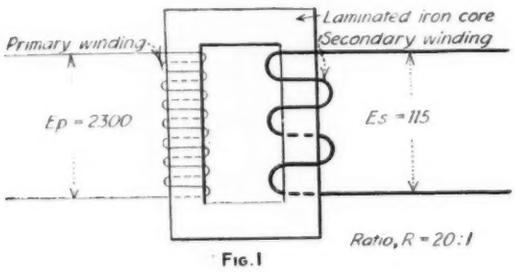


FIG. 1

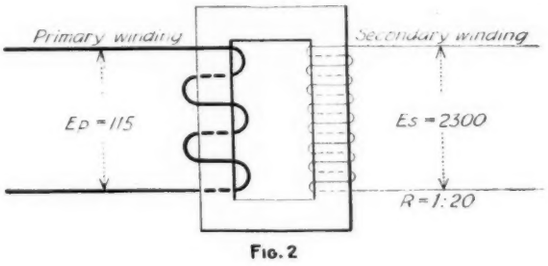


FIG. 2

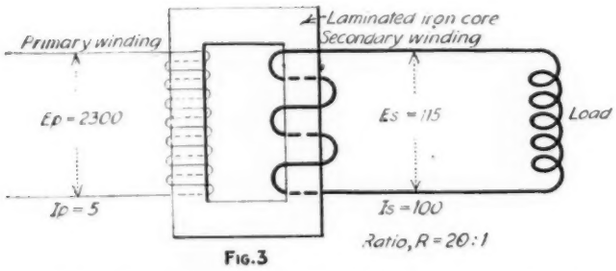


FIG. 3

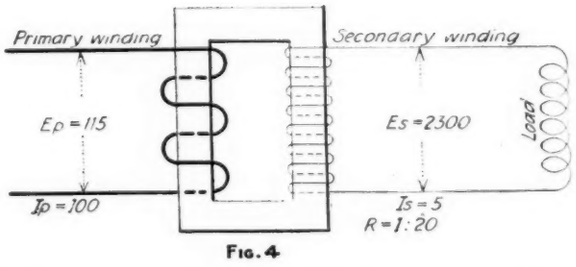


FIG. 4

FIGS. 1 TO 4—INDICATE THE RELATION BETWEEN PRIMARY AND SECONDARY VOLTS AND CURRENT IN STEP-DOWN AND STEP-UP TRANSFORMERS

consequently the current flowing with the secondary open-circuited is small. For the purpose of this discussion the induced voltage may be considered equal to the applied voltage, as would be the case in the theoretical transformer. That is, the alternating magnetic field set up in the core induces 2,300 volts in 2,300 turns. Now if another winding is put on the core, such as the secondary, with only $\frac{1}{10}$ of 2,300 turns, or 230, then this winding will have only $\frac{1}{10}$ of 2,300, or 230 volts induced in it, since the same magnetic field loops through this winding as through the primary. If the secondary winding has only $\frac{1}{20}$ the number of turns that are in the primary winding, then the secondary voltage would be only $\frac{1}{20}$ of the primary, or 2,300 ÷ 20 = 115 volts, which will be assumed to be the conditions in Fig. 1. Since the direction of the voltage induced in the primary winding is in opposition to the applied voltage, then the secondary voltage must be in opposition to the primary volts. consequently if a load is applied to the secondary the current flowing in this winding will be opposite to that in the primary winding.

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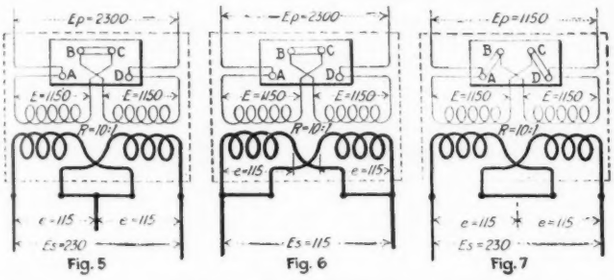
The ratio of the primary volts to the secondary volts is called the ratio of the transformer, which in Fig. 1 is $E_p : E_s$, or 2,300 : 115, and dividing each side by 115 gives a ratio of 20:1. If this were a step-up transformer as in Fig. 2, then with 115 applied to the low-voltage winding, 2,300 volts would be obtained from the high-voltage or secondary winding, and the ratio would still be $E_p : E_s$, or 115 : 2,300, and dividing each side by 115 gives a ratio of 1:20. In the first case the ratio



FIGS. 5 TO 7—DIFFERENT GROUPINGS OF THE PRIMARY AND THE SECONDARY WINDINGS

Fig. 5—Both the primary and the secondary windings are connected in series. Fig. 6—Primary windings connected in series; secondary windings connected in parallel. Fig. 7—Primary windings connected in parallel; secondary windings connected in series.

of the step-down transformer is 20:1, or $20 \div 1 = 20$, where in the latter with the step-up transformer the ratio is 1:20, or $1 \div 20 = \frac{1}{20}$.

The secondary volts E_s of a transformer are equal to the primary volts E_p divided by the ratio R, or $E_s = \frac{E_p}{R}$. This relation holds true whether the transformer is used to step the voltage up or step it down. In Fig. 1 $E_s = \frac{E_p}{R} = \frac{2{,}300}{20} = 115$, likewise in Fig. 2 $E_s = \frac{E_p}{R} = \frac{115}{\frac{1}{20}} = 2{,}300$ volts. It should be remembered that the ratio of a step-down transformer is a whole number, while that of a step-up is a fraction or a decimal, whichever may be used.

Relation Between Primary and Secondary Currents

The next thing to consider is the relation between the primary and the secondary current. Assume that the secondary winding is connected to a load that takes 100 amperes, as in Fig. 3. This current will flow in the secondary winding in opposition to that in the primary, therefore the secondary current will tend to demagnetize the core, whereas the primary current magnetizes it. It was assumed that the secondary had 115 turns, therefore with 100 amperes in this winding there will be $100 \times 115 = 11{,}500$ ampere-turns tending to demagnetize the core. As soon as the core starts to be demagnetized, the voltage induced in the primary winding is decreased, because this winding is now being cut by a smaller number of magnetic lines. Since the countervoltage in the primary is reduced, there is a greater effective voltage to cause current to flow, and the primary current increases in value. This increase is theoretically equal to a value that will add the same number of ampere-turns to the primary that are in the secondary. This causes one to neutralize the other, and the magnetism in the core remains at practically constant value. Since there are 20 times as many turns in the primary as in the secondary, the former will require only $\frac{1}{20}$ the amperes to have the same number of ampere-turns or, in this case, $100 \div 20 = 5$ amperes. This gives $5 \times 2{,}300 = 11{,}500$ ampere-turns in the primary, the same as in the secondary. From this it is seen that the primary current I_p equals the secondary current I_s divided by the ratio R, or $I_p = \frac{I_s}{R} = \frac{100}{20} = 5$ amperes. Conversely, the secondary current equals the primary current times the ratio, or $I_s = I_p R = 5 \times 20 = 100$ amperes. This also applies to the step-up transformer, Fig. 4, where, if the secondary current is 5 amperes the primary winding will take a current $I_p = \frac{I_s}{R} = \frac{5}{\frac{1}{20}} = 100$ amperes. If the primary current is 100 amperes, the secondary current is $I_s = I_p R = 100 \times \frac{1}{20} = 5$ amperes as given on the figure

Rating of Transformers

On either side of the transformer the volt-amperes equal the volts times the amperes. On the primary side Fig. 3 the volt-amperes $VA_p = E_p I_p = 2{,}300 \times 5 = 11{,}500$, and on the secondary side the volt-amperes $VA_s = E_s I_s = 115 \times 100 = 11{,}500$. This is for the theoretical transformer without losses. In practice the output obtained on the secondary side will be from 1 to 5 per cent less than the input on the primary side, depending on the efficiency of the transformer, which varies from 99 per cent for large well-designed units at about 75 per cent load and unity power factor to about 95 per cent for small units. The watts input and output are obtained by multiplying the volt-amperes by the power factor. In a transformer the power factor on the primary side will always be a little lower than on the secondary side, owing to the magnetizing current of the transformer being added to that of the load.

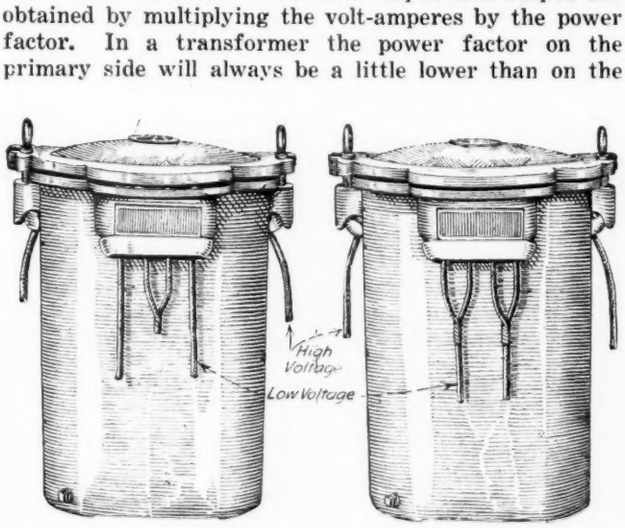


FIG. 8—TRANSFORMER WITH LOW-VOLTAGE WINDINGS CONNECTED IN SERIES

FIG. 9—TRANSFORMER WITH LOW-VOLTAGE WINDINGS CONNECTED IN PARALLEL

The rating of a transformer is based on its output on the secondary side. For example, a 50-kva. transformer will safely supply a load of 50 kw. at unity power factor continuously or a load of 40 kw. at 80 per cent power factor.

Most all small distribution transformers except some of those of modern design, have two primary and two secondary windings arranged so that they

may be connected in series or parallel. On the low-voltage side, which usually is the secondary, the two leads from each winding are brought to the outside of the case so that they are readily available for connecting, to obtain either full voltage, half voltage or both, as conditions may require. On the high-voltage side, which is usually the primary, the four leads are brought to a terminal block inside the transformer case. Figs. 5 to 7 show diagrammatically how the terminals are arranged and the possible voltages. In Fig. 5 the two sections of the primary winding are connected in series as are also the two sections of the secondary winding. With a ratio R of 10:1 between the primary and secondary windings, the secondary voltage Es will be $\frac{1}{10}$ of the primary, in this case 0.1 of 2,300 or 230 volts. This is obtained across the two sections of the secondary winding in series, so that one section will develop 115 volts. This makes it possible to bring out a third wire from between the two windings and have a three-wire system from which 115 or 230 volts may be obtained. In practice this is frequently done for a lighting load or a combined light and power load.

Primary Windings in Series, Secondary in Parallel

In Fig. 6 the primary windings are connected in series, as in Fig. 5, but the secondary windings are connected in parallel, so that only one-half the voltage can be obtained as was in Fig. 5—in this case one-half of 230, or 115 volts. However, double the current can be supplied from the secondary in Fig. 6 that could be in Fig. 5, so that the capacity of the transformer is the same in either case. The primary windings are connected in parallel in Fig. 7, and this makes the transformer good for only one-half the voltage on the high-voltage side that was applied to the windings when connected in series, or $2{,}300 \div 2 = 1{,}150$ volts, as shown in the figure. Since the ratio between the sections of the primary and secondary is 10 : 1, then with 1,150 volts applied to each section of the primary, each section of the secondary will have $1{,}150 \div 10 = 115$ volts, induced in it. With the two secondaries connected in series, $115 \times 2 = 230$ volts will be obtained, or a third wire may be brought out from between the two windings and a three-wire 115- and 230-volt system supplied as in Fig. 5. On the other hand, the two sections of the secondary winding may be connected in parallel as in Fig. 6 and a two-wire 115-volt circuit provided.

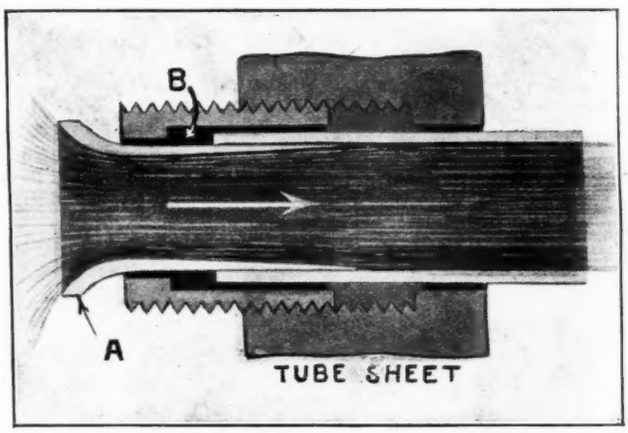
On the primary side the leads are brought to four terminals located equal distances apart. This allows using the same connectors for either the series or parallel connection. For series connection the two connectors are placed, one on top of the other, on terminals B and C as in Figs. 5 and 6. For a parallel connection one connector is placed from A to B and the other is placed from C to D, as in Fig. 7. This arrangement is not only convenient but eliminates the possibilities of making mistakes when changing from one connection to the other. On the secondary side the leads of the two sections are crossed inside the transformers, so that there is no crossing of leads on the outside of the transformer case when the two sections are connected in parallel. Figs. 8 and 9 show the secondary winding connected series and in parallel respectively on a distribution transformer.

In many of the earlier power systems 1,100 volts were used, and to make distribution transformers suitable for either 1,100 or 2,200 volts, arrangements were made inside the transformer for connecting the two sections in the primary winding in parallel or series, as described in the foregoing. Since 1,100-volt systems have practically gone out of existence, many of the modern distribution transformers have their primary winding connected permanently for 2,200 or 2,300 volts and no provisions made for a 1,100- or 1,150-volt connection. In some cases 5 and 10 per cent taps are provided on the primary winding so that the transformer ratio can be adjusted to obtain the proper secondary voltage, when the primary voltages vary 5 or 10 per cent from normal. However, the practice of bringing out four secondary leads is standard on most distribution transformers of 50-kva. capacity and less, for voltages below 6,000. On large power transformers various combinations of leads and taps are provided, depending upon the service requirements.

The next article will deal with the grouping of single-phase transformers for different voltage and phases.

Entrance Nozzle for Condenser Tubes

Among the many factors affecting the life of condenser tubes, general corrosion is probably the most common and can usually be attributed to one or a combination of causes. It is now generally conceded that



VENTURI NOZZLE PLACED AT INLET END OF TUBE

the entrained and dissolved air in the circulating water is responsible for much of the corrosion action and that the life of condenser tubes depends quite as much on the character of the water as on the properties of the tubes.

With a view to eliminating the corrosive action due to the gas pockets formed at the inlet end of the tubes (as at B) the nozzle A shown in the illustration has been developed. The nozzle, when placed in a tube as shown, tends to prevent the contraction of the jets of water as they enter the tube and thereby stop the liberation of oxygen at that point.

The design of the nozzle is such as to produce a venturi action and to increase the flow of water through the tubes.

The nozzles (known as Flowrites) are made in sizes to fit standard-diameter tubes and when in place form a practically continuous inside wall and provide a smooth entrance to the tube. They are manufactured and distributed by the Albert E. Mace Co., 755 Boylston St., Boston, Mass.

Steam-Trap Installation and Operation

OUTGROWTH of the writer's experience with steam traps, giving logical reasons for their erratic behavior and clarifying some of the misunderstandings and rule-of-thumb methods employed. Attention is given to functioning of traps, their location and need for venting.

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Chief Engineer, American Smelting & Refining Company

A STEAM TRAP may be defined as any device or apparatus designed and installed to remove automatically and continuously condensation or other water from a steam line without permitting the escape of steam and with the least possible heat loss. Traps may be divided into two general groups, continuous flow and intermittent. The general operating characteristics of all types of traps are independent of whether they are applied to high-pressure, low-pressure or vacuum service.

Perhaps the oldest type of trap, and certainly by all odds the most effective and satisfactory, is the water-sealed U-column illustrated in Fig. 1. Traps of this design are of the continuous-flow type. They never develop leaky valves, clogged discharge openings, collapsed floats or air binding, and may be depended upon to work continuously without repair or attention of any kind, over long periods of time. Their use, of course, is limited to low-pressure and vacuum service on account of the headroom required for the up-cast or pressure leg of the trap, this being approximately 2½ ft. long for each pound pressure difference between inlet- and discharge-valve pressures. To the writer's knowledge this form of trap has been employed successfully on the oil separators of open heaters, and a number of industrial heating systems where it has replaced expensive expansion traps and vacuum-heating drain valves.

Features of Operation

All types of traps other than the one already outlined, consist essentially of a container in which is a valve, automatically controlled, for permitting the passage of water and preventing the passage of steam. The principles underlying the operation of any trap are those governing the flow of liquids in closed containers or channels and may be summed up as follows:

1. Liquids acted upon by the force of gravity alone cannot flow uphill, but must always flow down.
2. To cause a liquid to flow uphill, it must be in a closed channel (pipe) and a pressure difference must exist between the bottom and top of the channel (pipe) equivalent to at least one pound per square inch for each 2½ ft. of water pressure head against which the liquid is to be raised.
3. Two bodies cannot occupy the same space at the same time.

A clear understanding that these three laws alone govern the action of steam traps, together with a clear perception as to how they apply is essential to the proper installation and satisfactory operation of steam traps.

Since the general function of a steam trap is that of removing a liquid from a vessel containing a gas, a study of the behavior of steam traps when applied to compressed-air lines may be of value as a preliminary to a study of their behavior when applied to steam lines.

Let us begin the study of traps by investigating an intermittent-discharge, single-valve trap of either tilting or bucket type, set above the point to be drained. Assume that this trap is to be installed to drain a water separator on an air line carrying 100 lb. air pressure, that it is set at an elevation of about three or four feet above the point to be drained and arranged somewhat as shown in Fig. 2. As to whether this trap will remove successfully the accumulated water from the separator may be determined by anyone who will procure a gallon glass jar fitted with a tight cork; insert tightly through this cork a short piece of pipe, and attach to the pipe a hose and funnel about three or four feet long. Support the funnel about six or eight inches above the jar and allow the hose to hang down below the jar in an easy loop. Now by pouring water into the funnel, fill the jar with water, as indicated in Fig. 3.

In this kindergarten illustration it will be seen that the glass jar represents the trap with a tightly closed discharge valve; the funnel, the receiving chamber of the separator; and the hose, the connecting pipe between the separator and the trap.

Why the Jar Will Not Fill

It is apparent that no amount of pressure (short of that necessary to burst the glass jar or blow out the cork) will cause water to flow from the funnel into the glass jar in sufficient quantity to fill it, because the system is "airbound" and in order to get the water into the trap the air must be allowed to escape or be "vented" off. It is therefore clear that a trap installed as in Fig. 2 could not be made to drain a receiver on an air line unless the discharge valve leaked, and then indifferently. This disproves the statement so often heard, "Oh, set the trap on the floor where you can get at it, as you have plenty of pressure to force the water up into it."

It is an authenticated fact that steam traps will work under these conditions. The main difference in operating conditions between traps installed on air and on steam systems lies in the fact that at all ordinary temperatures air remains a gas, while steam loses heat and condenses into a small fraction of the volume occupied.

Assume that the trap shown in Fig. 2 is installed on the separator of a steam line carrying 175 lb. absolute steam pressure. Several questions present themselves at once. Under what conditions will this trap operate? Is there any factor that will limit the speed of operation of this trap? Is it possible to figure approximately the minimum time between closing and opening of the trap?

In considering the first of these questions, it is apparent, from the conclusion arrived at for traps installed on air lines, that the trap cannot work unless care is used to blow out all air when starting it, and to provide means for removing accumulated air at frequent intervals thereafter. Almost all modern steam traps are provided with means for removing air from the trap

at each dump or discharge of the trap, although there are still a few in which this provision is omitted.

To both the second and third questions the answers can be arrived at simultaneously by a little figuring. Assume the cubical contents of the trap of Fig. 2 to be 1 cu.ft., the length of the trap body 18 in. and its diameter about 11½ in., both ends being assumed to be flat for simplicity of calculation. Suppose that the trap has just closed after a discharge, that it and the pipe leading to it are full of steam more or less wet and that a "slug" of about 1 cu.ft. of water enters the separator just after the trap closes. The condition would then be something like that shown in Fig. 4. The vertical pipe at the left would be full of water (assuming the pipe to be of ample size), and the horizontal leg would be

the loss of heat from a hot trap may be taken as equivalent to that of a steam radiator, or 1.7 B.t.u. per hour per square foot of surface per degree F. difference in temperature inside and outside the trap. Assume the room temperature surrounding the trap to be 80 deg., so that the difference between the inside and outside temperatures is 370.8 — 80 = 290.8 deg., or in round figures, 291 deg. The loss of heat per minute from the trap body will be (1.7 × 291 × 5.74) ÷ 60 = 47.3 B.t.u. The total units of heat resident as latent heat of evaporation in the 0.3843 lb. of steam contained in the trap is 0.3843 × 852.7 = 327 B.t.u. There will be condensed during each minute, 47.3 ÷ 327 = 0.145, or 14.5 per cent of the total steam in the trap, which is approximately equivalent to saying that

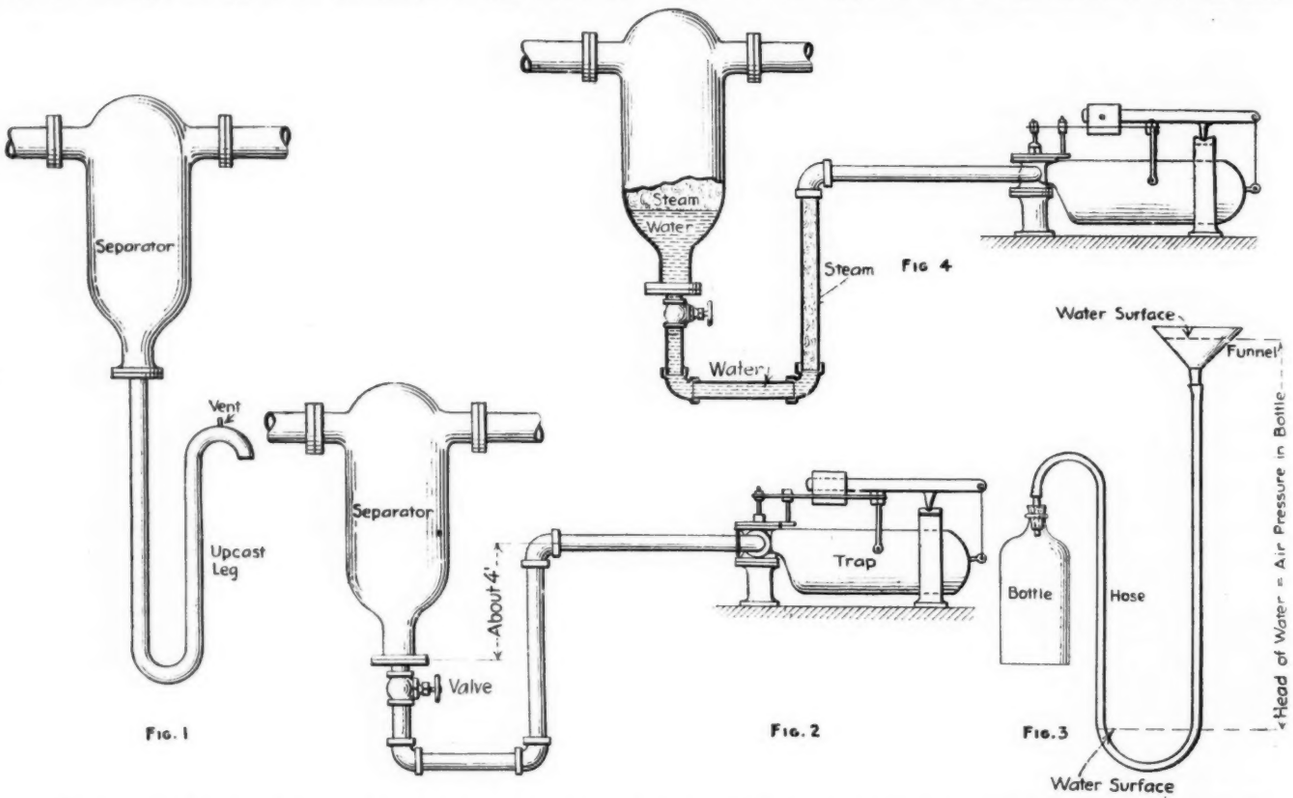


FIG. 1—WATER-SEALED U-COLUMN USED AS TRAP. FIG. 2—TRAP ON AIR LINE ABOVE POINT OF DRAINAGE. FIG. 3—OWING TO TRAPPED AIR BOTTLE DOES NOT FILL. FIG. 4—STEAM TRAP ABOVE THE POINT OF DRAINAGE

full of water, and the remainder of the pipe and trap would be full of steam at the same pressure as that in the separator.

Since the water cannot get into the trap because it is full of steam and the steam cannot get out because the water prevents its escape through the inlet line into the receiver, the only way the water can get into the trap is by a reduction in pressure in the trap due to the condensation of the steam, and it can get in no faster than the steam condenses.

It is now apparent that the time that must elapse before the trap can open to discharge this slug of water will depend entirely upon the rapidity of condensation of the steam within the trap body. This can be approximated in the following manner: The surface in the trap body approximates 822 sq.in., or 5.7 sq.ft. From the steam tables, steam at a pressure of 175 lb. absolute has a temperature of 370.8 deg., contains 852.7 B.t.u. per pound as the latent heat of evaporation, and weighs 0.3843 lb. per cu.ft. Under ordinary conditions

it will take over 4 minutes for 0.6 cu.ft. of water to enter the trap, which amount of water is about that required to dump a trap of the size assumed. Once the trap has opened, it will empty the separator and itself of water before it closes again, if the pipe line leading to the trap is sufficiently large to supply water as rapidly as the trap discharges it.

The answer to the second question is therefore that there is a factor limiting the speed of operation of steam traps set above the point to be drained, and that factor is the rapidity of condensation of steam within the trap body. To the third question the answer is that for any given set of conditions the minimum time interval between closing and opening of a steam trap set above the point to be drained can be figured approximately.

It is now apparent that a steam trap set above the point to be drained might be successful in removing water accumulating at a moderate and uniform rate, but would fail if called upon to handle large quantities of water accumulated suddenly, as when a boiler primes.

Investigation will be made now of conditions surrounding the operation of a trap set below the point to be drained. As before, assume that the trap is installed to drain a separator on an air line carrying 100 lb. pressure and that the arrangement is as shown in Fig. 5, this being by no means unusual as a trap connection. The question is, will this trap satisfactorily drain the air separator of accumulated moisture? Reverting to the glass bottle, funnel and hose, empty the funnel and hose of water, raise it to approximately the position shown in Fig. 6 and pour carefully into the funnel a small stream of water. If care is used and the stream of water is not too large, the glass bottle will fill with water quite readily, the air within the bottle making its escape up one side of the hose and through the funnel, as the water flows down the other side into the bottle.

Now empty the glass bottle of water, replace the cork tightly and arrange as shown in Fig. 6. Pour rapidly a large stream of water into the funnel. A considerable splashing and bubbling in the funnel will ensue, and if the hose be small enough, the flow of water into the bottle may stop entirely. In any event the bottle will fill more slowly than under the previous condition, because of the serious interference between the outflowing air and the inflowing water.

In view of the results obtained in the two simple experiments just performed, it may be concluded safely that under one set of conditions a trap connected as in Fig. 5 will remove satisfactorily the separated water from an air line and under another set of conditions the same trap may fail to function. A little further study of the results of the last two experiments discloses that the difference in performance in the two cases is due entirely to a difference in the venting out or removal of the air in the jar under the two conditions.

This suggests that if a way could be found to remove the air from the jar as rapidly as the water entered, and without interfering with the free flow of water into the jar, up to the capacity of the hose to carry water the more rapidly the water was poured into the funnel the more rapidly the jar would fill. Following this line of reasoning, if a second pipe, the same size as the one attached to the hose and funnel, be inserted tightly through the cork of the glass jar and left open, the jar will fill as rapidly as the hose and funnel will deliver water to it.

Applying these conclusions to the trap arrangement shown in Fig. 5, if a vent or pressure equalizer pipe be installed connecting the top of the trap with a point on the receiving chamber of the separator high enough so that it is not likely ever to be under water, an arrangement like Fig. 7 would be obtained, and it would be certain that the trap always would remove all water separated unless the quantity was entirely beyond the capacity that the trap could discharge.

Consider now the case of a trap connected as in Fig. 5 to the separator on a steam line. As pointed out previously, the only factor that can cause a difference in operation between the steam trap and the trap on the air line, when both are above the point to be drained, is the fact that steam condenses at ordinary temperatures while air does not, so that as long as the flow of water down the inlet line from the separator to the trap is not great enough to seal the pipe, the trap will operate as indicated in the first illustration in the discussion of Fig. 5. If either the flow of water or the piping condition were such as to trap the steam and

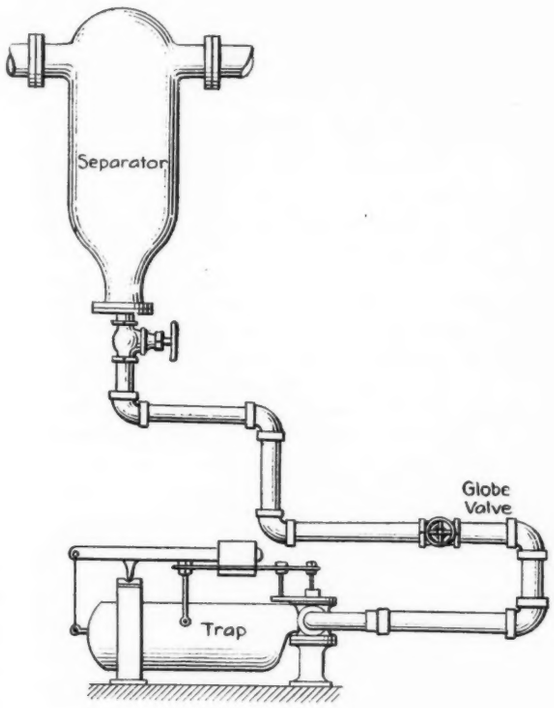


Fig. 5

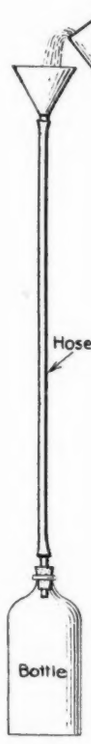


Fig. 6

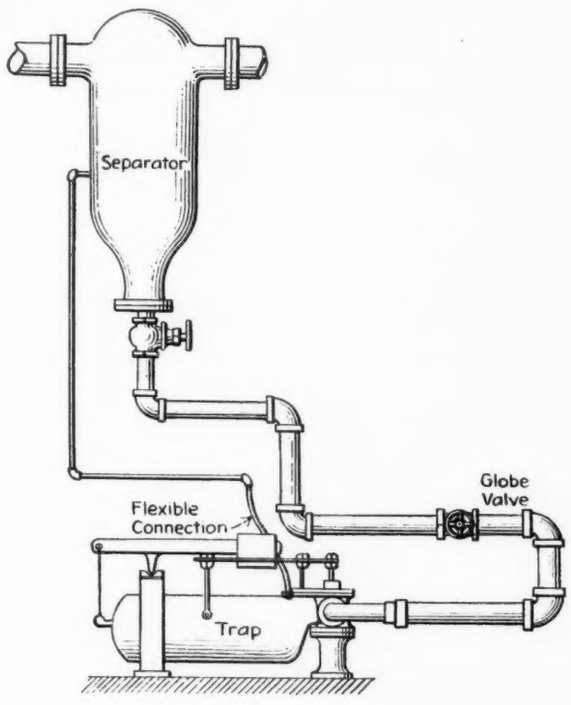


Fig. 7

FIG. 5—TRAP ON AIR LINE BELOW POINT TO BE DRAINED. FIG. 6—RATE OF FILLING DEPENDS ON QUANTITY OF WATER FLOWING DOWN HOSE. FIG. 7—STEAM TRAP BELOW POINT OF DRAINAGE AND VENTED TO SEPARATOR

prevent its easy escape up the inlet line as the trap fills, then it will operate exactly as does the steam trap placed above the separator, but possibly with a little shorter minimum time interval between closing and opening.

Precautions To Be Observed

From the foregoing it is seen that if a trap is expected to handle large quantities of water rapidly, the following precautions must be taken:

1 The trap invariably should be placed below the point to be drained.

2. The inlet line to the trap must be large in diameter, one to two sizes larger than the trap inlet, and as short and direct as possible.

3. All traps should have ample vent or balancer connections (see Fig. 7) so installed as to permit the maximum flow of water that the inlet line will carry, whenever such a flow is demanded by service conditions. This will lead to snappy trap action in emergency and may prevent a bad wreck in times of stress.

4. That all horizontal runs, right-angled turns or pockets may be avoided, the inlet connection to the trap should be preferably bent pipe with long easy turns, and a good pitch or drain from the separator to the trap inlet.

5. Globe valves in trap inlet lines should be avoided, but if they are used, they should be placed so that the seat of the valve does not form a trap or pocket in which water can collect and interfere with the escape of steam and air from the trap. Thus in horizontal or inclined runs of pipe the valve must be placed with the stem horizontal.

The writer is of the opinion that the need for venting (the installation of balancer or pressure equalizer piping) all classes of traps for all services, except traps placed above the point to be drained on steam services, is not widely realized by operating engineers, designing engineers or by trap manufacturers. It is his hope that this brief analysis of trap installation and operating conditions may excite real thought and discussion among all who have to do with installing and operating traps in power plants or elsewhere.

Northwestern Boiler Damper Control

In the Aug. 22, 1922, issue of *Power* a description was given of the Type B motor-operated damper regulator made by the Northwestern Manufacturing Co., of Milwaukee, Wis. As may be recalled, this was a large single unit, of which all the parts were mounted on a common bedplate and inclosed in a common housing. Owing to the fact that many boilers are arranged with narrow alleyways between them, making it inconvenient to mount the large machines in this location, the company has perfected a smaller machine, designated as Type G-2, which contains all the parts making up the single-unit machine, but for convenience and space saving they are separated into three principal units, known as the pressure unit, Fig. 1, the motor unit, Fig. 2, and the contactor cabinet, Fig. 3.

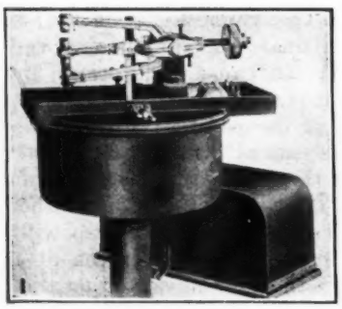
Operation of the control is as follows: The furnace pressure is admitted to the under side of a bell-shaped float, which is sealed off from the atmosphere with oil. The float is balanced by a weighted lever mounted on knife-edges and carrying electrical contacts on the end opposite the counterweight. Any change in furnace pressure tends to unbalance the weighted lever, moving it up or down to make contact with either the upper or the lower stationary contact shown in Fig. 1. The action closes a primary circuit to either the forward or reverse magnetically operated contactor, which in turn closes the motor circuit, causing the motor to run in such direction as to open or close the damper to compensate for a variation in pressure in the boiler furnace and to bring it back to normal. The control may be connected also to the stoker engine and to the draft fan if desired. If the furnace pressure should fall below normal, the damper will open, the stoker speed up and the fan run faster in order to correct the condition.

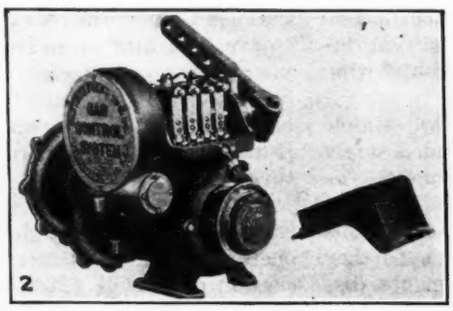
Fig. 2 shows the motor unit, which is fully inclosed and dirt-tight. It is built for direct current with dynamic braking to eliminate the possibilities of overtravel. The motor runs on ball bearings and develops about 350 ft.-lb. torque on the crank arm. Gears running in oil and mounted on ball bearings transmit the power developed by the motor to a drive shaft on which is mounted a cast-steel crank arm to which the damper is connected. A set of cam-operated limit switches mounted on the driveshaft can be adjusted so that the damper travel may be set for any desired arc up to 90 deg. The motor unit may be mounted on a bracket as close as convenient to the damper.

Fig. 3 shows the contactor cabinet containing the forward and reverse contactors, the main-line switch and protecting fuses. The cabinet is provided with a tight closing door and may be mounted on a wall or post in any convenient position.

Two adjustments control the operation of the regulator. Moving the counterweight on the float lever in or out will adjust for the pressure to be maintained. Turning two setscrews, one above and one below the weighted end of the balance lever, adjusts the amount of variation to be allowed.

In constructing a boiler to operate at extremely high pressures, it is necessary to make the drum a single thick-walled forging without joints.





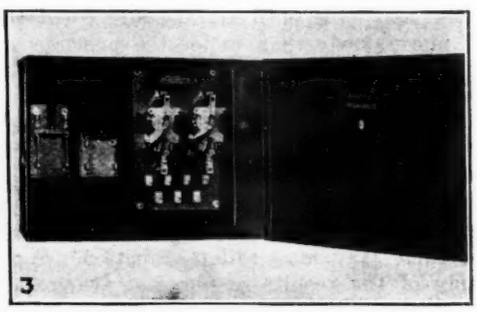


FIG. 1—PRESSURE UNIT WITH CASING REMOVED. FIG. 2—MOTOR UNIT. FIG. 3—CONTACTOR CABINET

EDITORIALS

F. R. LOW, EDITOR

Absence of Measuring Instruments Costs Money

THE total disregard paid by factory owners to their power plants is inexplicable in view of the high cost of coal burned under the boilers. The manufacturing end of the factory receives the closest scrutiny of the executives in an endeavor to here and there cut off a penny or so from the manufacturing costs. On the other hand, the power plant is considered to be merely a dark grimy hole into which is poured thousands of dollars each month. Just where these dollars go and how much is wasted is seldom given consideration by the executives. This probably is because it is hard to check the results obtained in the power plant, since the product, power, is consumed on the premises. Most industries find that five per cent of the cost of manufacture is traceable to the power plant. If the turnover is three per year, this represents fifteen per cent on the company's capital.

How would the executives receive a proposition to increase the dividend rate by seven per cent at little or no expense? Still this is possible by giving the plant engineer a little more consideration when he requests a few investments by which the coal consumption could be checked against the steam generation.

For example, in a coal survey conducted in Utica and Herkimer Counties, New York, it was found that only four plants had such measuring instruments. The other plants were being operated in hit-and-miss fashion, when a few instruments would have shown the engineers and owners how far from satisfactory were the plant results.

An engineer should refuse to handle a plant unless the owners show a willingness to assist him by purchasing instruments deemed necessary.

An "Outline" of Fuel

IN THESE days of "outlines"—outlines of history, of science and what not—it would not be out of place to have an outline of fuel. Perhaps we already have it in Bacon and Hamor's remarkable book, "American Fuels," but that book, while covering the field broadly, is a little too long and too complete for this purpose. For a real outline one may well turn to the masterful address "Fuel—Its Use and Abuse," which R. H. Fernald recently delivered before the Engineers' Club, in Philadelphia.

Picture, if you can, a cube of coal containing three and one-half trillion tons, the estimated total coal supply of this country before mining began. Imagine a smaller cube containing eighteen billion tons, cut from the corner. This smaller cube, one-half per cent of the total, is the coal mined to date in 111 years of mining. If mining should proceed at the same rate as the average since mining began, the rest of the cube would last twenty thousand years. At the 1917 rate, it would last four thousand years, while with consumption constantly accelerated at the present rate, the whole mass would be gone in one hundred years. Of course, the last is impossible because the rate of acceleration must eventually drop off, but somewhere between one hundred and four thousand years is the probable life of our coal supply.

In this way Mr. Fernald painted a striking picture of our coal resources and their rate of exhaustion. Then he drew another equally interesting picture of the distribution of the coal now being mined.

Imagine a train of twenty 50-ton coal cars at the mines. Before starting the train, 40 tons are dropped off at the mine for local use. Next, the railroads hold out 240 tons. The train then passes to the industrial centers, where it drops off 350 tons for the industrial steam trade and 170 tons for the domestic and small steam trades. Of the 200 tons left of the original trainload, 130 go to the coke manufacturers, 10 to the manufacture of coal gas, and 60 to steamship bunkers and export.

This is the distribution as pictured by Mr. Fernald. What savings does he consider to be reasonably possible under present conditions? The mines, he says, could save 12 tons out of 40. By partial electrification and better firing the railroads could save 120 tons out of 240. The industrial steam loads could save 79 tons out of 350, while the other consumers could save as follows: Domestic, 25 tons out of 170; coke, 13 tons out of 130; coal gas, practically none; export and bunker fuel, 15 tons out of 60. The grand total is 264 tons, or 26.4 per cent of the original trainload.

Space does not permit covering other points of almost equal interest, as Mr. Fernald considers the available resources, distribution of consumption, probable trend of consumption, possible savings, etc., for petroleum, natural gas and other fuels. There is only space to pick out one or two items that seem open to discussion. A heat-balance diagram for a typical isolated plant, as given by Mr. Fernald, shows only three per cent of the energy in the fuel actually used in manufacturing processes, after deducting three per cent for transmission losses and three per cent for friction in the manufacturing machines. This does not allow for the fact that many manufacturing plants have a large process-heating load, and all northern plants have an appreciable building-heating load. Again, it is stated that the burning of oil under stationary boilers should be discouraged, the supply being reserved for marine uses and internal-combustion engines. While this is perhaps to be desired from the national viewpoint of fuel conservation, it will be hard to prevent a man from burning fuel oil under boilers where the price is such that it pays him to do it. Moreover, is it not true that the consumption of gasoline by automobiles is the main factor determining the production of crude oil, and that this, in turn, due to the practical limitations of storage, throws the heavier oils on the market at such a price that the available supply can be sold in competition with other fuels.

Obtaining Information on Hydraulic-Turbine Operation

WHAT the development of the indicator by James Watt did for the steam engine, the oscillograph, as developed by A. E. Blondel and W. Duddell, made possible for electrical machinery and circuits. Previous to the invention of the indicator any opinion as to what took place in the cylinder of a steam engine was mere conjecture. By the application of the indicator it was possible to obtain a record of the change in pressure throughout the complete stroke, and what previously had been guesswork became recorded facts. It is not saying too much to credit the indicator with contributing more to the development of heat engines than any other single accessory. Today no engineer would consider the performance of an engine, either steam or internal-combustion, without taking indicator cards. In fact, the indicator has found a wide application on other machinery such as compressors, servo-motors on hydraulic turbines and others.

It is also safe to say that the oscillograph has contributed more to the knowledge of electrical machinery and circuits than any other device. On account of the cost of this apparatus and its being more complicated in construction and application, it will probably never come into as popular use as the steam-engine indicator. Nevertheless, new applications are continually being found for the oscillograph for recording both mechanical and electrical phenomena. In the former some means must be provided to change mechanical action into electrical in order that records may be obtained.

Two new applications of the oscillograph have recently been made to obtain a record of hydraulic-turbine operation. One of these was made by the Pacific Gas and Electric Company as reported in a paper, "A Study of Irregularity of Reaction in Francis Turbines," presented at the recent A.I.E.E. Pacific Coast Convention by Roy Wilkins. In this application a record of the varying pressures across the entrance to the turbine buckets was obtained. These records show at least one of the reasons for vibration in hydraulic turbines. Although the subject is new and little concerning it is known, this method opens up a new field of investigation in hydraulic-turbine design and operation.

Another new application of the oscillograph is described in the article, "Recording Hydraulic Turbine Operation with an Oscillograph," in the Sept. 18 issue, and in the leading article in this issue, "Results of Seventy-Three Tests made on Hydraulic-Turbine Governors." In these tests records of speed change, gate movement and load change or the pressure change in the penstocks were obtained simultaneously. In most of the tests a record of load change was substituted for the change in penstock pressure, as the latter in this particular plant was found to be of small consequence.

During the last twenty years more development work has been done in hydraulic turbines than in all preceding time. Manufacturers have built their own hydraulic laboratories in which extensive research work has been carried to completion; new and accurate methods of measuring the flow of water have been devised, improved designs of runners and draft tubes have been developed and the efficiency of the turbines brought to as near a state of perfection as can be expected; and the thrust-bearing problem on vertical units has been solved by two different designs. With all these improvements and others there has remained a need for some method of investigating the various phenomena taking place in these units during operation. In this respect the two applications of the oscillograph referred to suggest a possible means of obtaining information on many of the problems that have developed in the operation of the hydraulic turbine, which up to the present time have remained more or less a mystery.

Details Sometimes More Important Than "Essentials"

HIGH efficiency and reasonable cost, as consistent with economical conditions, are usually classed as essential factors of a power plant; facility of operation must necessarily be given sufficient attention.

It is one thing to provide a means of producing power at a financial gain and quite another to secure this production in a profitable manner. With the growth of large interconnected plants has developed an increasing appreciation of the value of service continuity. Means of obtaining very high efficiency at a reasonable investment would not succeed necessarily where service interruptions imperil the power market.

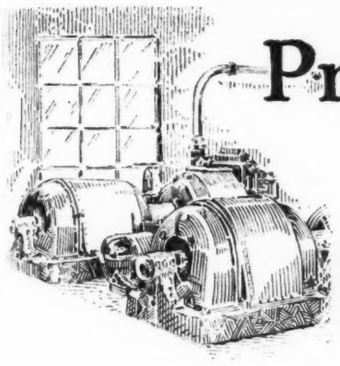
Efficiency, which is above the ordinary may be regarded as an essential; in practice, however, engineers are inclined to proceed with caution when means of greatly increasing efficiency are to be adopted; strictly speaking, it is not so essential as service continuity.

In studying the situation more closely, it will appear that a reasonable efficiency is absolutely necessary. Additional efficiency above this appears in the light of a desirable condition, but not so essential as reliability. It is notable that in one installation where steam pressures of about one thousand two hundred pounds are to be employed, the high-pressure division is operated to a large extent independently of the remainder of the plant. The main units are designed for the usual upper limits of pressure, representing reasonable efficiency as an essential. The unusually high pressure element, expanding to the main plant pressure, is a comparatively small feature with respect to relative capacity and is quite isolated in its location. Here it appears that considerations of the service continuity weigh more heavily than those of the additional efficiency.

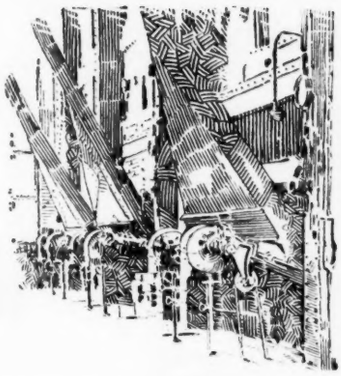
Valve-stem packing is seldom regarded as a vital part of a large prime mover. Generally speaking, it is not. As a matter of fact, it often happens that the life of the unit hinges on correct adjustment of the emergency valve stem packing. The emergency steam closing system is recognized as an essential, practically and economically. Details that appear inconsequential, yet may most easily get out of adjustment and thus nullify proper functioning, are actually of great importance.

Grouting in a steam turbine is essentially a dirty, uninspiring process. It is perhaps not generally realized that in many cases this is vitally concerned with permanent successful operation. It is sufficiently important to deserve more attention and analysis than have been generally extended so far.

It has recently developed that details of the boiler room were in reality so important financially that the boiler room has been referred to as offering a greater opportunity of saving money than any other part of a power plant. The operating man could doubtless define the essentials and details of a power plant as affecting economical power production, quite differently in many cases from those on which plans and purchases are based.



Practical Ideas *from* Practical Men



Solving the Problem of Distributing Power Costs

We have recently completed a six-story addition to our factory. Prior to two and one-half years ago we purchased all our power. With the addition of more space and equipment the question arose as to the advisability of continuing the purchase of power and operating low-pressure boilers for heating or to install high-pressure boilers and the necessary power units and generate our own power.

We estimated that with the addition of more space and equipment the exhaust from our power units would be practically all used for heating and process work during December, January and February, while during the remainder of the heating season part of the exhaust would be lost.

The question was: (A) Whether we would find it cheaper to operate our plant for heating purposes only and buy the power for the operation of our machinery, (B) find a use for a considerable portion of the exhaust steam in the non-heating season and operate the plant the year round, or (C) operate the power plant during the heating season, using the exhaust for heating, and purchase the power for operating the machinery during the non-heating season.

As we had for a number of years purchased our power and operated our boilers for heating, with the building of the addition we decided to adopt method (B) and installed a new, complete plant and took out the central-station service.

All glue machines, glue tanks and embossing presses, which formerly were heated by gas or electricity, were converted to steam heat, and exhaust steam is used, thus making a better load factor for our plant, as the electric load is thereby reduced, and in the non-heating season there is a considerable demand for exhaust steam. The returns from these heaters are taken care of through vacuum traps and pump, and practically no back pressure is carried on the engines.

It is possible, by keeping an accurate check on the demands for power and heat, to determine at what point it would be profitable to purchase the power over and above the requirements for steam heating from the exhaust.

All water fed to the boilers is metered and the makeup, which passes through a water softener, is also metered. At certain outside temperatures all the exhaust steam is used for heating. As the outside temperature rises, some of the exhaust is lost to the atmosphere.

We have found that from month to month there is a ratio between the makeup and the total boiler feed. As an example, for the month of January 87 per cent of the exhaust steam was returned for boiler feeding and 13 per cent was made up. During March 85 per cent was returned and 15 per cent was made up. In May 25 per cent of the exhaust was recovered and 75 per cent was lost. During June, July and August practically all the exhaust was lost except that used for process and heating feed water.

To arrive at a definite month-to-month cost for heat and power, we totaled the fixed charges and divided by twelve to get the monthly charge. To this is added the cost of wages, coal, water, removal of ashes, supplies and other sundry items of the current month.

If we did not use our power equipment, it would be necessary to use the boilers for heating, so we charge that portion of the monthly expense to heating in proportion to the demand on the exhaust steam. In other words, in January 87 per cent of the monthly expense was charged to heating and 13 per cent to power. In March 85 per cent was charged to heating and 15 per cent to power. In June, July and August the total expense is charged to power. The power units are charged directly to power the year through, as they are strictly a power item.

We have found this method of spreading the costs quite satisfactory in our case. We should, however, welcome any practical criticism from other readers.

Kansas City, Mo. H. N. CEDARLAND.

High-Speed Engine Cylinder Lubrication

In a department of the State of Texas there are fourteen high-speed engines. The question of cylinder lubrication was up for decision. Not finding anything of a definite character in any technical publication to guide us, we have, from our own records, produced a chart that can be used to advantage by engineers.

As most engines are equipped with quart lubricators,

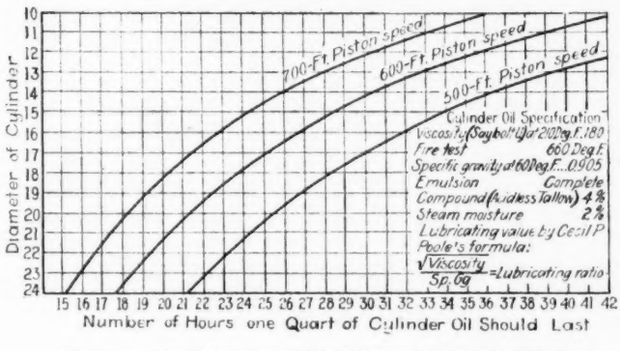


FIG. 1—AMOUNT OF CYLINDER LUBRICATION

this is used as a basis, the lubricator filled and the feed is set so that the quart will last the time shown in the chart.

In case it is desired to test the efficiency of the rate of feeding, the engine is stopped at the end of the stated time and the cylinder head removed. Three thicknesses of white cigarette paper are placed without delay on the cylinder walls. After one minute they are

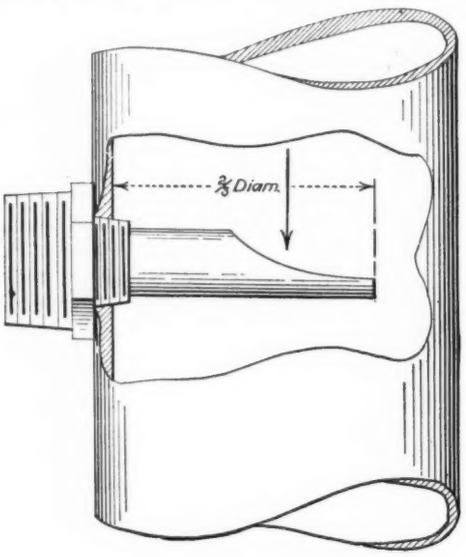


FIG. 2—LOCATION OF ATOMIZER

removed, and if the oil has penetrated through two of the papers, the lubrication is satisfactory.

It is well to know the average percentage of moisture in the steam, which can be determined by a calorimeter test. For each 1 per cent of moisture, 2 per cent of compound should be used in the oil. The cylinder oil should have the specification as shown on the chart for saturated steam of not over 150 lb. pressure.

In selecting such oils, it is well to use Cecil P. Poole's formula, which is given on the chart. We find it to be an excellent guide in deciding on cylinder oils. The chart is based on two million square feet of rubbing surface per pint of lubricating oil with an atomizer of the type shown in Fig. 2, placed 24 in. above the throttle valve.

J. P. GREENWOOD.

Austin, Tex.

Fitting a Clutch in Close Quarters

The starting air is a vital factor in a gas-engine plant, and means should be provided to assure a plentiful supply and to be sure that no emergency can arise that the supply cannot be readily renewed in case it is lost. One case that may be of interest came to my attention recently.

In the plant in which the incident occurred, were two belted motor-driven compressors, the motors taking current from the main busbars on the switchboard. Means had also been provided, whereby one of the compressors could be belted to a small gas engine, by aligning both the motor and the engine with the compressor driving pulley, but owing to the fact that breakdown service was available, it was evident that the small engine had never been used for the purpose.

There came a time when the breakdown service was discontinued, and one morning in trying to get the main units started, the engineer exhausted all the air without getting either of them on.

The small gas engine was fitted with a pulley that ran so close to the wall that the belt could not be put on after the engine was started, and as there were only two men on shift, they could not roll the engine fast enough to get it started with the belt on and pulling the compressor. Consequently, all the traffic on about one hundred miles of electric railway had to wait until the day crew came and helped get the small engine started with the belt on and until the air was again pumped up.

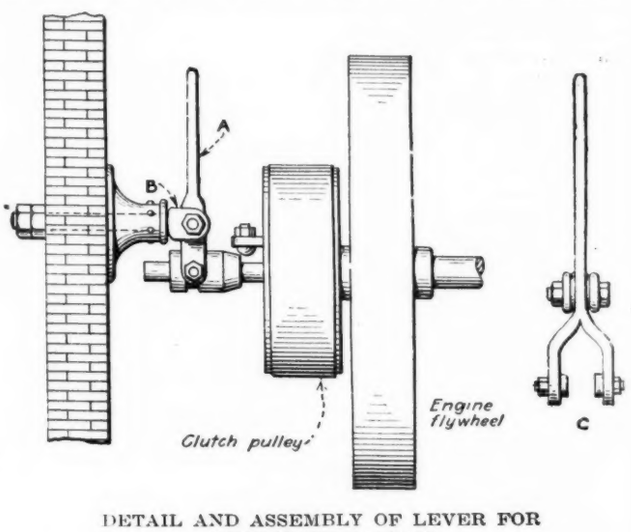
To avoid a repetition of this trouble, it was decided to get a clutch pulley and put it on the engine so that the belt could be kept on the pulleys and the compressor started at any time without delay.

One was found locally, which could be made to answer by drilling bolt holes in three of the engine flywheel arms to receive the clutch bolts. A distance piece was also required between the wheel and the clutch flange, to hold it out far enough so that the belt would clear the flywheel.

There was a handwheel on the clutch to engage and disengage it, but there was not enough room between it and the wall for a man to throw it in and out, so other means had to be provided.

An old brake reach rod from a freight car was found around the plant and one end was cut off to act as a handle. The other end was cut the right length to go through the wall to act as a support or pivot for the handle. An old Ford hub was also found, which was just the right length to hold the pivot or support the right distance from the wall.

The forked end of the piece used for the shifting lever *A* was spread just enough to fit over the clutch



DETAIL AND ASSEMBLY OF LEVER FOR OPERATING CLUTCH

spool, and the heads of the capscrews were turned off to fit into the groove in the clutch spool and were then fastened in the lever as shown in the detail at *C*.

The handwheel was entirely removed and after drilling a hole through the wall the lever and the support were assembled as shown.

This arrangement made it convenient to start the engine without load, and a man could stand behind the belt wheel, reach over and throw the clutch in with one hand.

EARL PAGETT.

Independence, Kan.

Comments from Readers

Straightening Copper Floats with the Aid of a Vacuum

Some time has elapsed since the issue of July 17, in which Fred E. Cupper discusses an article by R. A. Hall in the May 29 issue on the "Straightening of Copper Floats," but perhaps the writer's experience along this line will carry a hint for others.

In common with a good many of his "buddies," the writer chose to take his discharge from the army over in France, finding employment with a certain "Tuyauterie et Plomberie" (Pipes and Plumbing) manufacturing sheet-metal parts and plumbing supplies. One of the first tasks met in the capacity as production foreman was the restoration of several hundred copper spheres ranging from 5½ to 7 in. in diameter. Common report had it that the floats had been destined for use with U-boat nets; but whatever their original purpose, they had been so badly damaged in transit that there was none in the lot that was not "bashed in" at least once, and some of them showed four or more spots measuring up to three inches across and reducing the buoyancy to such an extent as to make them practically worthless as floats.

Our first attempt was by the time-honored method of soldering a wire loop in the middle of the depression and then attempting to pull it back into place. It worked, after a fashion, but bade fair to cost more than we could hope to recover by the sale of the floats.

We then tried inflation with compressed air, but found that the time for making a pressure-tight connection was equal to that for soldering the wire loops and that the pressure necessary to restore the spherical shape ran up to 60 lb. per square inch, at which point a number of the floats split or exploded, finally injuring one man so seriously that the boss stopped the work, saying he'd take his loss on the floats and sell them for scrap copper. That put it up to the one American in the shop to show the natives that Yankee resourcefulness was not wholly stumped.

Early in the job we had found, by puncturing one float under water, that they were filled with air at about atmospheric pressure, and it was decided to give this imprisoned air a chance to do the work, simply by reducing the pressure on the outside surface. For this we took a small cast-iron cylinder, closed at one end, and faced off the other end to an arc approximating that of an average float. A gasket of soft, well-oiled leather was then fitted over the open end, covering the face-off portion and extending down over the outside of the cylinder walls far enough to be fastened in position with a clamp. The side wall was drilled and tapped for a ¼-in. pipe, and a short lead run over to a sink, where the simplest kind of a vacuum pump was rigged up by carrying the small pipe through a tee into a ¾-in. pipe, turning it, and extending it for some ten inches inside the larger pipe. The ¾-in. pipe was then fastened vertically, so that water could flow down through it and around the ¼-in. nozzle.

The cylinder was bolted to a bench, open end up, and a sphere nested snugly, under hand pressure, into the leather gasket. Water was then allowed to run through the improvised pump and the air within the cylinder exhausted. Presently we were rewarded by hearing a faint "tung-g" as the dished spot sprang back into place.

The cost on the whole job was that of machining the cylinder and the labor of changing floats from time to time; while the balls restored in this manner were not strained, their seams were unharmed, the original shape was regained, and the lot brought in a nice profit when sold.

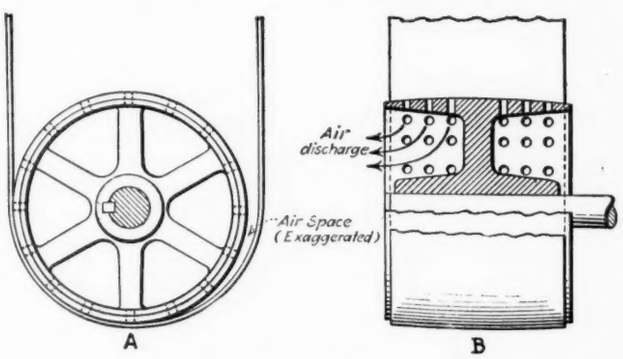
CHESTER LYNNDELLE.

Trenton, N. J.

Drilled Holes in Pulley Prevent Belt Slippage

The article by G. A. Luers in the June 19 issue is a case where the conditions of good belting practice were not properly met.

Drilling holes in pulleys to prevent slippage is by no means new. Some years ago in a shop where I was employed there was a 12-in. belt driving a heavy shaft



AIR IS DISCHARGED THROUGH HOLES IN PULLEY FACE

about 20 ft. overhead. This belt gave a lot of trouble through slipping, and on watching it closely, a wedge-shaped space could be observed between the belt and the pulley on the driving side, as shown at *A* in the illustration. This space must have been at least one-quarter of the theoretical arc of contact. This trouble went on for some time and the remedy tried consisted in tightening the belt and liberal application of belt dressing.

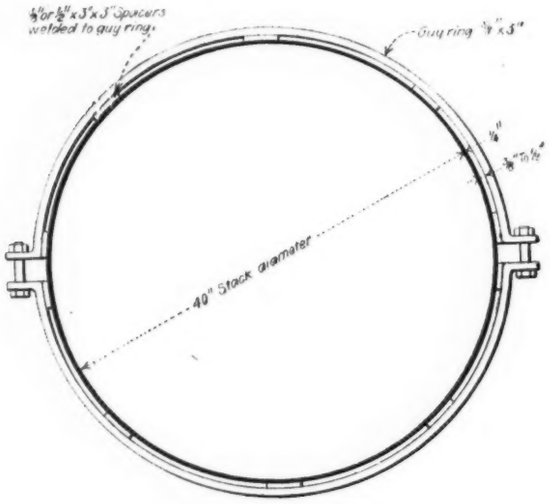
The trouble was finally overcome by drilling a number of small holes in the pulley. The effect of this was to cause the belt to hug the pulley more closely on the driving side. A discharge of air was now noticed as indicated by the arrows.

Turning grooves in pulleys to release the air is good practice but by no means new. The advantages of the holes over the grooves is that they do not seem to require cleaning out as the grooves usually do.

Winnipeg, Man., Canada. O. PETTERSSON.

Factors That Affect the Life of Iron Stacks

I read with interest the article by J. O. Benefiel in the issue of Aug. 28, "Factors That Affect the Life of Iron Stacks." This subject is important to many engineers and Mr. Benefiel's remarks are quite true, especially as to driving up riveted joints cold. This is poor practice. The upper sections always give way before the lower. However, after handling a number of steel stacks during the last twelve years, it is my opinion that the failure is due in a large measure to the manner in which the steel guy bands are placed around the stack. From some fifteen stacks dismantled I have found that invariably the section of stack to which the guy bands were clamped was rusted through or nearly so where the bands went around the stack.



GUY RING IS HELD AWAY FROM STACK BY SPACERS TO LESSEN CORROSION

The reason for a stack failing at the point mentioned is that the band is usually pulled to place in such a manner as to leave openings where water or condensation can gather together with fine ash, etc., between the band and section of the stack. This starts a rusting action, and in a short time the action of the wind and water weakens the stack at this point and it begins to show signs of distress. In order to overcome this trouble I tried the method of attaching the guy ring shown in the illustration, which after a two-year trial apparently is worth the additional cost and trouble of applying guy bands.

Referring to the sketch it will be seen that the band, instead of coming in direct contact with the stack, is held away at least $\frac{3}{8}$ in. by sections the width of the band and 3 in. long, welded to the band. These spacers allow water, fine ash, etc., to pass freely.

From my experience with steel stacks I recommend the thickness of sheet for a stack 40 in. diameter by 80 ft. high to be $\frac{1}{4}$ in.; this size stack to be assembled in four sections of 20 ft. each; joints to be riveted with not less than $\frac{1}{2}$-in. diameter rivets driven hot; stack to be given before erection or assembly a thorough brushing and then one coat of red lead and one coat of first-grade stack paint. After assembly the stack should be given an additional coat of stack paint.

The matter of properly guying a steel stack should be given serious thought as it usually receives little attention and guys are often placed in perhaps the most convenient place, but such places are often not correct when we consider the effect it will eventually have on the stack. Guys placed where they will actually do their work will relieve a great many stresses on the riveted joints and add materially to the life of the stack.

Cumberland, Md. PAUL R. DUFFEY.

Reminiscences of a Veteran Mechanical Engineer

I have been much interested in the reminiscences, by W. H. Odell, published recently in *Power*; especially those relating to the late George H. Corliss. Also the description of an old timer by M. V. White in the Aug. 28 issue, recalls some of my earlier experiences with this type of engine.

About 1912 I was called to Ayer Junction, Mass., where there was installed an old Corliss engine. I was told it had been on its present foundation forty years and was second-hand when moved there. In some manner the valve rigging, which was attached to the frame between the cylinder and rocker arm had broken, and the parts had been sent to our shop to be duplicated, and I was sent later to assemble them.

The engine had the general appearance of the girder type of today except the valve rigging, which was operated by heavy leaf springs about three feet long, set vertically, and as I remember, did not have a wrist-plate. There were long rods running to the valve stems,

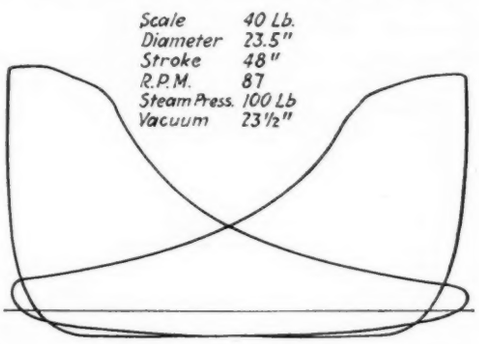


DIAGRAM OBTAINED FROM CORLISS ENGINE NO. 105 AFTER TWENTY-FOUR YEARS OF SERVICE

the crab-claw arrangement being on the frame, with the dashpots on the floor the same as now, with exhaust motion positive. I do not recall the exact size, but I think it was about 12x36 in. and run at 51 r.p.m. I indicated the engine at the time the repairs were made and obtained just as fine a diagram in every way as from a present-day engine. Unfortunately, I neglected to retain a copy of the diagram. If I remember correctly, the name-plate on the engine read Nightingale & Corliss and was dated 1852.

The diagram shown herewith was obtained from a George H. Corliss engine No. 105, which ran in the Merrick Thread Company's plant at Holyoke, Mass., for twenty years and was moved to the J. S. Viles Electric Plant, Middlesex, Vt., in 1900. This card was taken after the engine had been on the new foundation about four years, and used only as a standby in connection with a hydro station. This engine, I understand, is still there but not in use, as a modern steam plant has been built to take care of low-water periods.

Bloomfield, N. J. E. A. EARLE.

Catalogs and Advertising for the Boys

The letter of Mr. Terry in the Aug. 14 issue entitled "Catalogs and Advertising for the Boys," impressed me strongly. Well I remember the days of my youth, when to look through the door of an engine room meant more to me than a circus parade.

My father ran a small engine in a planing mill, and there is where I got my first idea of steam. I was in the engine room constantly whether it was mother's wish or not, and many a time I was punished for being there.

When I became older, I made friends with all the engineers in town. There were six steam plants in the village, the largest of which was rated at 250 hp. These engineers, some of whom have long since passed on, were my idols. They took an interest in me and helped me along. One man especially, familiarly known as Hi, used to let me do almost anything from filling the lubricator to starting the big (?) 200-hp. engine. To him I owe a debt that money cannot pay. He taught me how to use the indicator, how to key up a box, how to set a valve and the thousand and one things around a steam plant that it is necessary to know, in order to operate the plant successfully.

Another man that played a big part in my early life was the school superintendent. He was never too busy to answer a question or help me with a problem. When it came to the mathematical side of the business, the engineers were not very proficient, although they realized the importance of it and impressed on my mind the fact that mathematics was the foundation on which to build if I ever intended to get very far in the engineering profession.

There were also several salesmen visiting the plants, who were friendly to "the kid," as I was called. They gave me advertising matter and instruction books covering their products. It is needless to say that the kind treatment given me by these salesmen was bread cast upon the waters.

The modern plant of today is sadly lacking in the pleasant atmosphere that was found in the days gone by. There is very little done now, in this part of the country at least, to encourage the boy to take up steam engineering. Did you ever stop to think, Mr. Chief Engineer, that perhaps a potential engineer is bringing your paper to you every morning, and how a kind word of reference to the subject might be the means of starting him in the right direction. Do you meet the youngster who pauses to look in, with a smile and a kind word? I am afraid not many do. Do you advise for or against the calling when asked for advice?

And you, Mr. Salesman, when you meet the young fellow in the engine room do you give him a smile and a word of encouragement along with some advertising matter when he shows he is interested? This is certainly one way to sow seeds in good ground that will bear fruit tenfold.

So let us take stock of ourselves and see if we have done all we could to encourage the younger generation to enter a calling that is a long way from being obsolete (as is supposed by many) and one that carries with it as much real satisfaction and pleasure as any other vocation a man might follow.

E. W. Brong,
Chief Engineer, Power Plant,
North Canton, Ohio. Hoover Suction Sweeper Co.

Air Pressure in Surface Condensers

In an article entitled "Determining Efficient Operation and Causes of Low Vacuum in Surface Condensers," in the Sept. 11 issue S. Thomas suggested that the difference between the referred vacuum and that corresponding to the temperature of the discharged circulating water be taken as a measure of the partial air pressure in the condenser. The assumption is made that the temperature of the discharge cooling water corresponds to the temperature of the vapor in the condenser or to the vapor pressure. When it is considered how small a quantity this air pressure is, even in surface condensers, in which the leakage is great, the error in Mr. Thomas' assumption will be apparent.

Consider a 10,000-kw. turbine with a water rate of 12 lb. steam per kilowatt-hour and an air leakage of 10 cu.ft. of free air per minute, which would be considered large for a turbine of this size. Assume that 2 cu.ft. of free air per minute is carried over with the steam and that the vacuum in the condenser is 29 in. mercury referred. Twelve cubic feet of free air per minute would be equal to approximately 0.85 lb. per minute with a steam flow of 120,000 lb. per hour, and a final quality of 90, there will be 1,180,000 cu.ft. of steam per minute passing through the condenser neck and entering the tube bank. With an air leakage of one pound per minute the air density outside of the tube bank will be 0.000,000,85 lb. per cu.ft. The unreasonableness of trying to measure air pressure of the magnitude corresponding to this air density is at once apparent.

The best indication of this effect of air blanketing upon the surface-condenser heat transfer can be obtained by applying Dalton's law to the air vapor mixture in the pipe leading to the air pump or ejectors. As the steam passes through the banks of tubes toward the air-pipe suction, the air density increases until the air-pipe suction is reached. Here the density of the air must be such that the density times the volume removed will equal the weight of air leaking into the condenser.

If the air leakage increases and the volumetric capacity of the air ejector or pump remains the same, the absolute pressure and therefore the density of the air entering the air pipe must increase. In my opinion as the air pressure increases, the vapor pressure must decrease a corresponding amount. The reduced vapor temperature as well as the blanketing of the air causes the heat transfer to be very low in the portion of the tube bank surrounding the air-removal pipe.

The effect of the air upon the vacuum in a surface condenser is determined by the size of this zone of high air pressure. The size of this zone of high air pressure is determined by the air density or pressure in the air and vapor mixture entering the removal pipe.

The air pressure in the air-pump suction can best be determined by taking temperature and pressure measurements at the same point in the pipe. Care must be taken in securing accurate measurements since with tight surface condensers and efficient air pumps, the air pressure will not exceed 0.3 in. mercury. In cases where the air pipe is surrounded by steam pipes, the vapor mixed with air in the air-pump suction will be found superheated. In this case the air pressure cannot be determined.

J. Dare McAdoo.
Parnassus, Pa.

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Size of Main for Exhaust Steam Heating

What size of pipe should be used for main to supply 2,400 sq.ft. of heating surface with exhaust steam? T. J. B.

To meet all conditions, it would be well to have the supply, at the first point of distribution, of a size appropriate for a low-pressure gravity return system, which should have a diameter in inches of about the square root of $\frac{1}{100}$ of the gross number of square feet of heating surface, inclusive of risers, branches and returns, as well as wall coils, radiators or other forms of heating surfaces. For 2,400 sq. ft. the supply pipe should have a diameter of $\sqrt{\frac{2,400}{100}}$ = about 5 in. diameter. From the engine to the first point of distribution the pipe size should be larger, according to the length and number of elbows.

Sizes of Pump Suction and Discharge Pipes

What should be the size of suction and discharge pipes of a duplex pump? W. C.

For ordinary lengths of suction and discharge pipes the sizes should be such that the velocity in the suction pipe will not exceed 240 ft. per min., and that in the discharge pipe 300 ft. per min. Conservative manufacturers usually provide pumps with connections suitable for pipe sizes that are adequate for these velocities when the pumps are operated at their normal capacities, and to insure good performance of a pump without undue loss of power in pipe friction, the pipe sizes should conform to the velocities stated or be at least as large as provided in the connections. As it is customary to quote pump capacities in gallons handled per minute, the pipe diameters can be expressed by the following:

Suction diameter in inches $= \sqrt{0.1\,G}$

Discharge diameter in inches $= \sqrt{0.008\,G}$

where G = number of gallons handled per minute.

Shortest Cutoff Practicable for Throttle-Governed D Slide-Valve Engine

What are the considerations that limit the shortness of cutoff of a throttled D slide-valve engine, and what is the least fraction of cutoff practicable? W. L. R.

With a D slide valve an earlier point of cutoff is obtained by advancing the eccentric and adding lap to the valve so the steam edges of the valve will not uncover the ports too early before the beginning of a stroke of the piston. Movement of the valve is quickest when the eccentric is in its highest or lowest position and must be slower as the dead-center position of the eccentric is approached. Hence, by advancing the eccentric, the slower is the opening and also smaller the port opening, varying from the most rapid opening and widest opening of the steam ports, when there is no angular advance, to no opening during admission stroke of the piston when the eccentric is set in line with the crank. Hence with a moderate amount of valve travel and width of valve, to obtain early cutoff, the port opening will be so slow and so small that the cylinder pressure realized at the beginning of the stroke and average pressure during admission will be much below the steam-chest pressure. Advancing the eccentric also makes the release earlier and results in too much compression for a throttle-governed engine. If the release is delayed by adding lap on the exhaust edges of the valve, the exhaust closes earlier, further increasing the compression and reducing the mean effective pressure. For these reasons it is not practicable to operate a throttled slide-valve engine with cutoff shorter than about $\frac{5}{8}$ of the stroke.

Low CO_2 and no CO in Flue Gases

If 6½ per cent of CO_2 and no CO is found in the flue gas from a boiler with forced draft, what is wrong and what should be done to remedy the trouble? P. S.

With no CO there should be 12 to 15 per cent of CO_2. The low percentage of CO_2 would indicate good combustion of the coal that is burned, but signifies that there is large excess of air entering the passes for the gases along with the products of combustion. This might result from having holes in the fire, areas of the grate where no combustion takes place, air leaks directly from the ashpit to the combustion chamber, or, where there is a strong stack draft in addition to forced draft, there may be cracks or other openings that admit air along with the gases from the furnace before reaching the point where the gas sample is taken for analysis. The remedy would be to carry a clean active fire over the whole grate and stop all air leaks.

Maximum Allowable Working Pressure

What determines the maximum allowable working pressure for a new horizontal return-tubular boiler? P. C. B.

The maximum working pressure depends on the highest pressure that is safe for the kind, strength, dimensions and arrangement of the material of the shell, rivets, stays and tubes and the character of the workmanship. The strength of the shell usually is taken as the criterion of safe working pressure. The maximum allowable working pressure on the shell is determined by the strength of the weakest course, computed from the thickness of the plate, the tensile strength of the plate,

the efficiency of the longitudinal joint, the inside diameter of the course and allowed factor of safety, computed by the formula,

$$P = \frac{TS \times t \times E}{R \times FS}$$

where

P = Maximum allowable working pressure, lb. per sq.in.;
TS = Ultimate tensile strength of shell plates, lb. per sq.in. of cross-section;
t = Thickness of shell plate, in weakest course, in.;
E = Efficiency of longitudinal joint, or ratio of strength of the joint per unit of length to the strength of solid plate for the same length of shell;
R = Inside radius of the weakest course of the shell, in.;
FS = The allowed factor of safety.

Brine Temperature for Ice Plant

What should be the temperature of the brine for freezing ice, using condensed steam water and what back pressure should be carried to obtain that temperature? G. H.

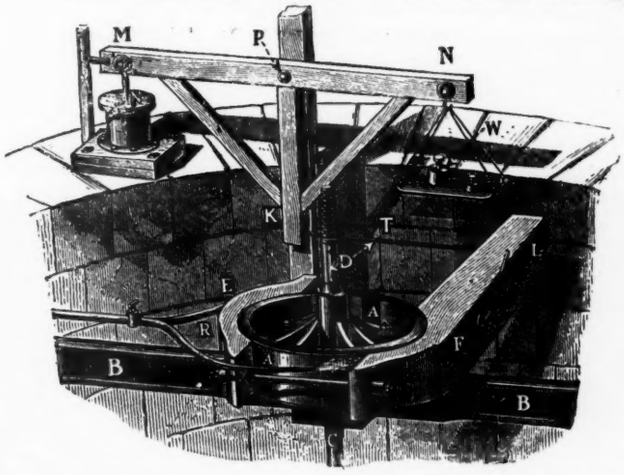
The brine temperature should be 12 to 14 deg. F. and there should be a difference of from 12 to 14 deg. F. in temperature between the brine and the ammonia in the coils of the brine tank. In the *Power* booklet entitled "Refrigerating Pointers," there is a chart that shows the correct ammonia temperature for each brine temperature, and for 12 deg. the temperature of the ammonia should be —5 deg. F. The pressure corresponding to this ammonia temperature is practically 10 lb. gage, and that is about the pressure that should be carried. It is quite possible to carry the ammonia temperature slightly higher than —5 deg. so that the pressure can be slightly above 10 lb., but it will require more coil surface than needed with the —5 deg. temperature, or the temperature of the brine will go up. There must be a certain amount of heat absorbed by each square foot of coil surface, and the differential between the ammonia temperature and the brine temperature depends on the surface of the evaporating coils, the temperature of the brine and the temperature of the ammonia.

Testing Vertical Shaft Turbine Waterwheel

How can we test the power developed by a vertical shaft turbine waterwheel? D. F. M.

For testing the power of a moderate-sized vertical shaft waterwheel the energy developed is absorbed and measured by means of a Prony friction brake, placed on a horizontal brake wheel carried by a vertical shaft coupled to the wheel, with the brake arm resisted in the horizontal plane of rotation of the brake wheel. In order that the effort exerted may be taken up and weighed, or balanced by a dead weight, the horizontal tangential pull or thrust of the brake arm must be converted to the vertical direction by a bell-crank, or equivalent motion, operating in the same vertical plane.

The general arrangement of a brake for testing power obtainable from a vertical-shaft turbine with rotation clockwise is shown in the figure. A heavy double flanged cast-iron brake wheel A is keyed on the vertical turbine shaft CD near a shaft bearing carried by the bridge tree B. A brake shoe E and brake lever FL are made of maple, or other even-grained wood, carefully fitted to the face of the brake wheel and clamped to it by two strong bolts one of which should be provided with a nut easily accessible for adjustment of friction from clampage of the brake shoes on the face of the wheel. The brake arm FL is connected by the rod KL to the vertical arm KP of a double bell crank pivoted at P and providel with horizontal arms PM and PN. From one horizontal arm of the bell crank the scale pan is hung for receiving scale weights W, and the other is connected to the loose-fitting piston of a dashpot for controlling vibrations of the brake arm. The brake wheel is lubricated by water jets from the forked pipe R. All parts should have enough strength and rigidity to resist about four times the torque exerted by the shaft for, with almost any form of brake, there is likely to be violent chattering due to eccentricity of the brake wheel from imperfect construction, setting



ARRANGEMENT OF PRONY BRAKE FOR TESTING POWER OF VERTICAL SHAFT WATER TURBINE

and irregular expansion from the heat generated, vibrations of the shaft, elasticity of the brake arm and bell crank and un-uniform lubrication between the wheel and brake shoes.

For computing the power developed, the length of brake arm to be taken is the perpendicular or least distance DT in feet from the center of the shaft to the center of the rod KL, and with the bell-crank arms balanced on the pivot P, the force acting at the end of the brake arm will be the weight in the scale pan, in pounds, multiplied by the length of the arm PN, divided by the length of the arm PK. For operation of the brake, gradually adjust the opening of the waterwheel gates, obtain the normal r.p.m., or speed of test, by adjusting the nuts of the brake shoe bolts, and note the number of pounds balanced in the scale pan when the speed is maintained for a considerable time. The level of head and tail water should be simultaneously marked by assistants. Calling W the number of pounds balanced in the scale pan then, the number of brake horsepower developed would be

$$\frac{(DT \text{ in feet} \times 2 \times 3.1416) \times r.p.m. (W \text{ lb.} \times PN \div PK)}{33{,}000}$$

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Easy Lessons in Engineering

Grate, Radiation and Leakage Losses in Boilers

Almost invariably the largest loss in boiler operation is that due to the sensible heat in the stack gases. This loss, as explained in the easy lessons of the Sept. 25 and Oct. 2 issues, is easily figured from the flue temperature and the percentage of CO_2. The other losses mentioned in these articles were that due to incomplete combustion and that due to the evaporation of moisture. The incomplete-combustion loss may be of serious proportions, but it can be almost completely prevented if the operator is on his guard and makes occasional Orsat tests for CO. The loss due to the evaporation of moisture is not generally of great importance. Moreover, it is entirely outside the control of the fireman, since he cannot possibly burn the coal without evaporating the moisture.

In addition to these flue losses there are other boiler losses, all of which *should* be of a minor nature, although careless operation may cause them to reach serious proportions. These are loss of combustible through the grate, radiation loss from boiler settings and bare steam drums and losses due to leaks of steam and hot water.

The loss of combustible through the grate, while not ordinarily large in comparison with the flue loss, is still worthy of careful attention. If the ashpit refuse contains a high percentage of unburned fuel, it is thereby made evident that the grate, the stoker or the method of handling the fire is not well suited to the fuel used. The remedy is largely a matter of applying common sense to prevent good coal from falling into the ashpit.

It may be well occasionally to determine this grate loss by an actual test to supplement the approximate information obtained by the inspection of the refuse. The figures obtained, particularly if reduced to the basis of tons of coal per year or dollars per year, will usually be high enough to impress the fireman with the importance of keeping the ashpit loss down to a minimum without neglecting the more important matter of maintaining the proper percentage of CO_2.

If the total weight of ash and refuse handled per day is known, and a test is made of a fair sample to find the percentage of combustible matter contained, a simple multiplication will give the total weight of combustible matter lost per day through the grate.

The first step is to get the fair sample. To do this, shovel out several hundred pounds of ashpit refuse on a clean concrete floor. Break up the larger clinkers with a sledge. Then mix thoroughly with a shovel and divide the pile into four quarters. Discard the opposite quarters and crush the remaining ash a little finer. Then mix, quarter and discard as before. Repeat these operations several times until the sample has been reduced to five or ten pounds. If this sample is not perfectly dry, it should be air dried for several days in a shallow pan placed in a warm place, say on top of a boiler. Then weigh the dried sample carefully. The next step is to burn out all the combustible matter. This can be accomplished by keeping the sample at a dull red heat for several hours while exposed to the air. One way to do this is to place the sample in a short open length of large pipe heated in a forge. The sample is then cooled and weighed again, the loss of weight being the combustible matter contained.

As an example, assume that the dried sample weighs 10 lb. and that the weight after burning out the combustible is 6.7 lb. Then the weight of combustible is $10 - 6.7 = 3.3$ lb., or the fraction $\frac{3.3}{10} = 0.33$, or 33 per cent of the total weight of the ash and refuse. If, say, 4,200 lb. of refuse is produced each day by the entire plant, the total daily loss of combustible will be $0.33 \times 4{,}200 = 1{,}386$ lb. This loss is greater than that of 1,386 lb. of coal because coal contains non-combustible matter—ash and moisture. Assume that the proximate analysis of the coal used is as follows: Moisture, 2.0 per cent; volatile, 30.6 per cent; fixed carbon, 58.2 per cent; and ash, 9.2 per cent. Then the percentage of combustible matter in the coal is $30.6 + 58.2 = 88.8$ per cent, so that 0.888 lb. of combustible is equivalent to one pound of coal. Therefore the 1,386 lb. of combustible is equivalent to $1{,}386 \div 0.888 = 1{,}560$ lb. of coal, which may therefore be taken as the daily ashpit loss for the plant.

Another method of computation will give the loss directly as a *percentage* of the coal fired without knowing the total weight of refuse produced per day. For example, in the case just figured, a 10-lb. sample of refuse shows 3.3 lb. of combustible and 6.7 lb. of true ash, so the combustible is $\frac{3.3}{6.7}$, or 49.2 per cent, of the true ash. When one pound of coal is fired, the true ash passing to the ashpit must be 0.092 lb. (from the coal analysis); so the weight of combustible lost per pound of coal must be $0.092 \times 0.492 = 0.0452$. Since, as previously figured for this fuel, 0.888 lb. of combustible is as good as one pound of coal, the coal equivalent of one pound of combustible will be $0.0452 \div 0.888 = 0.051$ lb. In other words, the grate loss is 5.1 per cent of the coal fired.

Heat losses due to steam and hot water wasted through leaky safety and blowoff valves and frequent popping of safety valves, are easily detected and remedied. That due to radiation from exposed boiler drums should be reduced to a minimum by suitable heat insulation. The heat loss through the setting may be reduced through the use of insulating brick, but this should be applied only under expert guidance if refractory troubles are to be avoided.

High-Pressure Steam Boilers

Among the papers of particular interest presented at the seventeenth annual convention[1] of the Iron and Steel Electrical Engineers was that of Dr. D. S. Jacobus on high-pressure steam boilers. As an introduction Dr. Jacobus traced the history of high-pressure boilers from the time, over one hundred years ago, when Jacob Perkins built an apparatus to generate steam at several thousand pounds pressure.

He then described experiments made by the Babcock & Wilcox Co. about eight years ago with a boiler having a

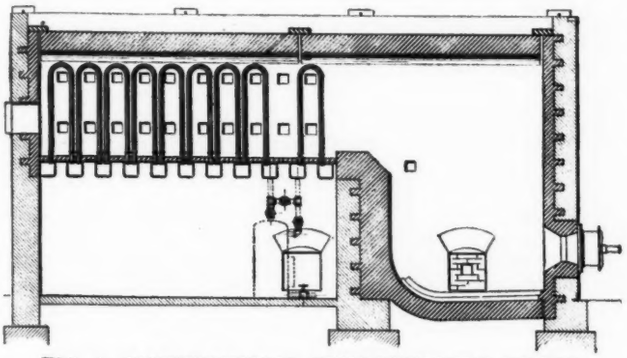


FIG. 1—EXPERIMENTAL HIGH-PRESSURE BOILER DESIGNED FOR "SERIES" OPERATION

forced circulation and with no steam or water drum. This boiler (shown in Fig. 1) was purposely made of a form in which it was felt there might be certain difficulties, in order to search out the principles involved in such boilers. It was called a "series" boiler, because the water was forced continuously through a series of loops. Starting at the back, the first loops acted as an integral economizer, and the other loops, up to the separating tank (shown in dotted lines), as actual evaporating surfaces. The steam and water from the next to the last loop passed to the separating tank, from which the saturated steam rose to the final superheating loop next to the fire. The loops were made of 2-in. tubes with cores in the straight sections.

This boiler was operated at a maximum capacity of about 425 per cent of rating, based on the entire amount of heating surface of the economizer, boiler and superheater. The surface was not arranged in a way to give efficient results, the object of the test being to determine the features of circulation and the practicability of operating a boiler of this type.

Since this boiler had no water space and hence required 100 per cent reliability in the feed-pump system, it was decided to build a series boiler to be used in combination with a relatively small boiler having a steam and water drum. The installation is shown in Fig. 2.

This consisted of a boiler with 2-in. tubes having a steam-and-water drum and a combined boiler and economizer with 1-in. tubes, in which the heating surface was 3.7 times the heating surface of the boiler having a steam and water drum. For simplicity these will be called the upper and lower boilers respectively. There was 187 sq.ft. of tube surface of the upper boiler, 51 sq.ft. of superheater surface, and 685 sq.ft. of boiler and economizer surface in the lower boiler, or 923 sq.ft. in all. The boiler was oil-fired and operated at 600-lb. gage pressure. A connection fitted with a valve was run from the steam and water drum to the lowermost part of the lower boiler, and this connection was left open in starting up so as to provide a local circulation. The unit was run, at times, at over 500 per cent of rating, based on the total surface, and all parts were found to operate in an entirely satisfactory way. The upper boiler operated at ratings as high as 1,500 per cent.

Experiments made with this boiler led to the final design of the 1,200-lb. boilers now being built for the Weymouth and Calumet stations. Since this type of boiler has already been described in *Power*,[2] we may pass on to the following digest of Dr. Jacobus' comments on high-pressure boilers.

If it were possible to maintain absolutely tight surface condensers, it would be unnecessary to clean the interior of the boiler tubes and the problem of designing high-pressure boilers for power-plant service would be greatly simplified. Under present conditions there is always a chance of leakage at the condensers, so the problem both of cleaning the interior of the tubes and of removing any excess oxygen from the feed water must be faced in all cases.

The effect of temperature on the lowering of the elastic limit of steel has been referred to by engineers as something that might give trouble in a high-pressure steam boiler. At a temperature of about 750 deg. F. the elastic limit may drop to half that obtained at ordinary room temperatures, and it is important that the effect of this falling off be understood. The temperature of the saturated steam corresponding to 1,200 lb. pressure is somewhat less than 570 deg. F., and at this point the strength of the steel is higher than at ordinary room temperatures and the elastic limit is about the same or only slightly lower, so there need be no apprehension with respect to the boiler proper. The only part that is affected through being brought to a temperature that would cause any material difference in the strength or the elastic limit of the steel is the superheater. Of all parts of a boiler the superheater is the safest as far as its liability to injure anyone or to cause property damage is concerned, as the worst that might happen in case the superheater became overheated would be to burn out or

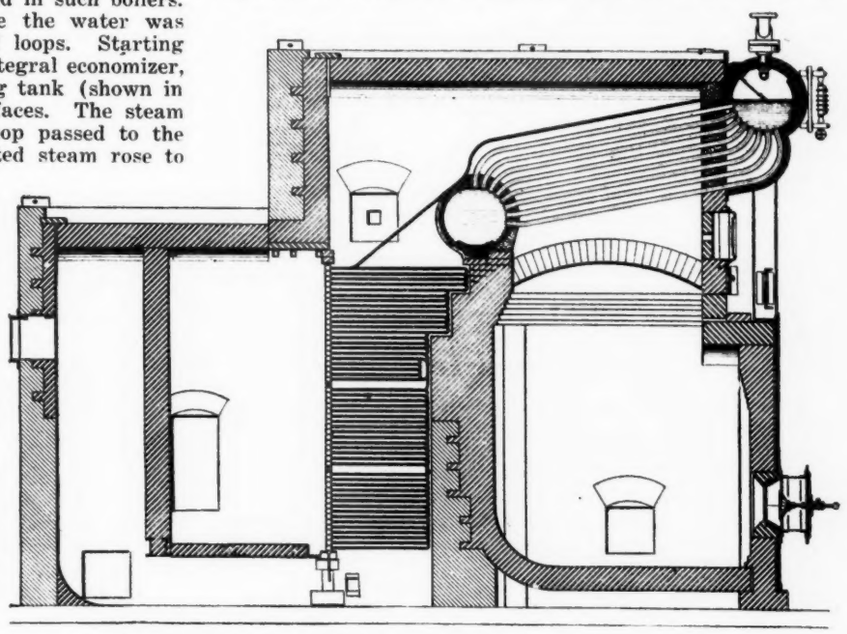


FIG. 2—LATER FORM OF HIGH-PRESSURE EXPERIMENTAL BOILER

blow out some of the tubes, and this would not ordinarily result in any damage other than to the superheater itself.

The effect of a rising temperature is first to increase the tensile strength of steel, say, 10 to 20 per cent, and afterward to decrease it. At 700 deg. F. the tensile strength is about the same as at ordinary room temperatures. At 750 deg. the tensile strength falls off, say, 10 per cent below what it is at ordinary temperature, and the elastic limit is about one-half what it is at ordinary room temperatures. In a complicated structure used under pressure at 750 deg. F., the falling off in the elastic limit would result in a liability to leakage, and in such a structure it might be well

[1] Held Sept. 24 to 28, 1923 at Buffalo, N. Y.

[2] On May 29 *Power* described the 1,200-lb. boiler at Calumet, while the July 10 issue described a similar boiler at the Weymouth station.

to use a factor of more than 5 based on the tensile strength.

In a structure like a superheater, however, there would seem to be no necessity of providing a factor of safety of more than 5 based on the tensile strength. In boiler construction we have become so accustomed to the use of factors of safety that are relatively high in comparison with those used in some other lines that we are prone to overdo matters to make sure we are on the safe side. When one considers the relatively low factors used in the design of large guns and torpedo tubes carrying very high pressures, one feels a great deal safer when it comes to boiler design. Again, in the work we have done in designing and building oil stills and tanks for use at high pressures and a temperature of 825 deg. F., we made a careful study of the falling off of strength and elastic limit and of the practice in still design that has been found to give safe results, and all this makes us more confident in feeling that the factors of safety we are using in boiler and superheater design are ample.

In the latest boilers for high-pressure and high-temperature superheated steam built by the Babcock & Wilcox Co., the superheater is placed above a series of tubes in a drop-leg and below the main body of tubes of the boiler. A superheater properly arranged in this position in a properly designed boiler gives a more constant degree of superheat at different ratings than if placed in the ordinary position above the boiler tubes, and permits a higher degree of superheat to be obtained with a given amount of superheating surface. Moreover, for the temperature of superheated steam now employed, the superheat obtained with a superheater of the sort is near enough constant for use without an attemperating device.

There is but little gain in economy in increasing the steam pressure to a higher point than 350 lb. per sq.in. without interstage heating of the steam; that is, withdrawing the steam from one stage, resuperheating it and returning it to the following stage. Interstage heating necessarily involves additional complication and added expense, and the possible difficulties in operation due to the additional complication have an important bearing on the use of the higher steam pressures.

Since the foregoing was written, a new development in the previously mentioned type of 1,200-lb. boiler has been announced. This consists in moving the resuperheater from its original position enveloping the primary superheater in the space above the drop leg to a position above all the tubes. The purpose of this proposed change is to make possible the accurate regulation of reheat temperature by controlling the flow of gases past the reheater.

Emergency Power for Mines

In coal mines that produce considerable gas, it is essential that power be available at all times for operating the ventilating fans and the hoist, if the mine is of the shaft type. Now that central-station power is being used so extensively for mine service, the continuity of service in gaseous mines is very important. Although the central station is reliable, trouble may occur on the transmission lines.

Some mines have kept their steam equipment intact and ready for operation in case the purchased power should fail. This is an expensive procedure, for the boiler plant must be fired at all times.

In a number of instances gasoline-engine-driven generators have been installed to furnish emergency power for the service hoist. As most service hoists are of the alternating-current type, the generator of the gasoline-engine-driven set is usually of the alternating-current type and is arranged to drive the service hoist motor at full speed, or an extra motor that can be thrown into gear on either the main or service hoist is employed and the hoist operated at a reduced speed. This arrangement is used because large gasoline-engine units are expensive.

An alternative arrangement was described recently by Graham Bright in a paper before the Canadian meeting of the American Institute of Mining and Metallurgical Engineers. In this the author has developed a plan by which the main hoist and the fan can both be operated at reduced speeds, in case of loss of power from a central-station power system. In this plan a gasoline-engine-driven generator operates at a reduced frequency, if alternating current, and a reduced voltage if on a direct-current system. With alternating current the engine speed and generator are arranged to give a frequency of approximately one-half the normal frequency, but this can be varied to meet the particular conditions at any mine. In case the generator operates at 30 cycles, the voltage will be 1,100 if the hoist and fan are operating on 2,200 volts 60 cycles. The motors would then operate at one-half speed with the same control and would give a fair power factor and efficiency, for the voltage has been reduced in the same ratio as the frequency. In operating the main hoist, in many cases, the load of men will be about one-half of the load of coal or ore. As, in addition, the speed is reduced one-half, the power required will be about one-fourth of that necessary when hoisting coal or ore.

Stopping Corrosion with Silicate of Soda

That corrosion, at least in hot-water supply lines, can be prevented not only by removing the oxygen, but also by the use of special "dopes," particularly silicate of soda, was an interesting point brought out by C. R. Texter, in an article published in the September issue of the *Journal* of the American Water Works Association. Mr. Texter who, as an Industrial Fellow of the Mellon Institute of Industrial Research of the University of Pittsburgh, is making a special study of corrosion, briefly outlined the prevailing theory and showed how, with normal water, corrosion can proceed only as the dissolved oxygen in the water removes the hydrogen film that would otherwise protect the surface. He then stated that while corrosion could be prevented by the use of deaeration or deactivating apparatus, it was sometimes possible to produce the same result by coating the interior of the pipe or tank with a protecting film.

In some cases this film is formed naturally. For example, plants that use the Great lakes water have little trouble with corrosion, because the calcium carbonate, which this water contains in large quantities, is thrown out in the hot-water supply lines as a hard scale, which protects the surfaces. In New York City, on the other hand, the water, being unusually soft, furnishes no such protection, so that hot-water supply lines there are often rusted through in two or three years.

Experiments have been made in the past to determine to what extent artificial protective films may be effectively used to prevent corrosion. Water pipes have been coated with zinc, aluminum, cement, and a variety of asphalt and coal-tar preparations. Any of these must be applied to the pipe before it is put into service. Once the water has eaten through the protective material there is no chance of its renewal, and the bare steel is exposed to the destructive action of water. This shortcoming of most artificial coatings suggested the desirability of a protective coating that can be applied continuously or intermittently after the pipe has been put into service in the hot-water system. Search for a suitable protective film was started about three years ago in the research department of the National Tube Co., and for the last year has been continued by the Mellon Institute, working mainly with silicate of soda (water glass), which the preliminary experiment showed to be best suited to the purpose.

The value of silicate of soda lies in the fact that it is readily carried in solution by the hot water throughout the system and is precipitated (by combining with colloidal iron or soluble salts in the water) in the form of an impervious film which prevents the corrosive water from coming into actual contact with the metal. One important advantage of this process is that the treatment may be made continuous or intermittent as it is found necessary, the color of the water being a good indication of the control. Other alkalis, such as lime or caustic soda, have been tried, but they are less permanent and the film is more easily penetrated by the water. With silicate of soda it was found possible to cut off the supply for two or three

weeks and still keep the water clear. The process has been used for over two years in Pittsburgh and for something over a year in New York and Boston, with considerable success. The treatment has been found applicable to old rusty hot-water heaters and almost worn-out pipe lines, as well as to new installations.

On the other hand, the silicate process is said not to be practicable for very long lines, because in such cases the protective material is mainly deposited in the first sections, leaving the more distant sections unprotected. In the writer's opinion the main use of the silicate treatment will be in laundries, small apartment houses and private dwellings. He does not state to what extent, if any, the treatment could be successfully employed in power plants. Further information on this point would be of interest.

Novel Pump Primer

All engineers who have had experience with centrifugal pumps that are situated much above the level of the water they have to lift, are well aware of the difficulties that often occur in starting. Water will have leaked away through the foot valve and left the casing full of air, which has either to be displaced by priming water or extracted by an ejector before the pump can be got to work at all. Whenever possible, of course, centrifugal pumps should work "drowned," but there are many cases in which this cannot be arranged, and under such circumstances the Seaborne interceptor described in *Engineering* is likely to prove a most useful piece of apparatus. It enables a centrifugal or other pump to be started at once without preliminary priming or other attention, no matter how great the suction lift may be within reasonable limits, so long as the suction pipe is airtight. It is, moreover, free from any moving parts and of such extreme simplicity that it does not appear to contain any possibility of trouble.

Essentially, the Seaborne interceptor consists of a closed vessel of cast iron or steel plate placed on the suction line

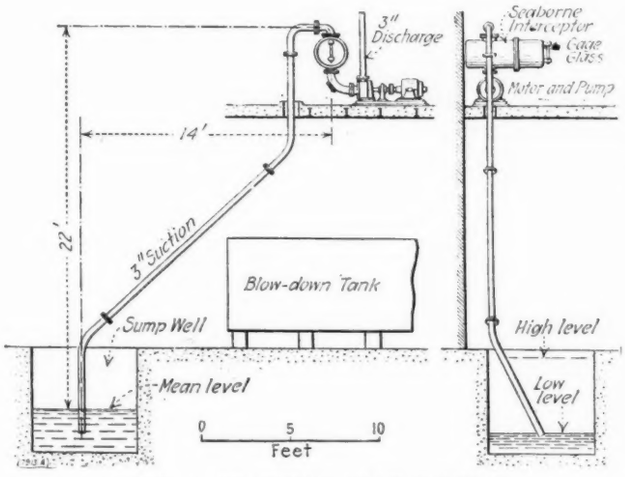


FIG. 1—SEABORNE PUMP PRIMER

of the pump and at a somewhat higher level than the latter. The inlet to the interceptor is at the top, and the outlet to the pump is directly beneath it. Between the two is fixed a conical pipe as shown in Fig. 2. The position of the interceptor relatively to the pump is shown in Fig. 1. The action is as follows: when starting up, the upper end of the discharge pipe and the whole of the suction pipe may be filled with air, the only water in the system being that in the pump and interceptor casing, which, together with a short length of the discharge pipe, form a kind of "trap." When the pump is started, it takes water from the interceptor, lowering the water level in the latter and creating a partial vacuum in the suction pipe. This, of course, causes water to rise in the suction pipe, and the interceptor is so proportioned that water will start flowing into it at the top before it has been emptied by the pump. The interceptor will at this time be nearly full of the air that was in the suction pipe.

The incoming water passes down to the outlet through the short conical pipe, and in so doing acts as an ejector, taking with it some of the air through which it passes before entering the cone. In a short time practically all the air that was originally in the suction pipe will have been carried through the pump by this means, and the interceptor will be nearly full of water. This is the normal state of affairs when running. When the pump is stopped, the water will run back through it until the level in the discharge pipe has fallen to that of the top of the interceptor

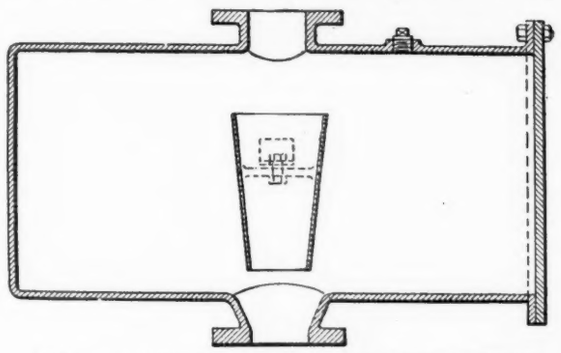


FIG. 2—INJECTOR NOZZLE AND INTERCEPTOR

The remaining air in the latter is sufficient to prevent any siphoning action, so that the vacuum is broken and the suction pipe empties itself, leaving the pump and interceptor both full of water. The pump can then be restarted at any moment, the interceptor acting in the manner already described.

The arrangement shown in Fig. 1 represents an actual installation of the interceptor in one of the London power stations. It will be noticed that there is a vertical suction lift of 22 ft. to be overcome and that no foot valve of any kind is fitted. The duty of the pump is to empty a dirty water sump at frequent intervals, discharging the water into an overhead tank. We have been given the opportunity of noting the behavior of the interceptor in actual work, and in every instance starting was effected as readily as if the pump had been drowned. A gage glass fitted to the interceptor casing enabled the action to be watched. In addition to its primary function of facilitating the starting of pumps, it is clear that the interceptor can also embody a strainer to prevent injurious materials from causing damage to the pump. The makers of the device are the Seaborne Interceptor and Engineering Co., Ltd., of 45, Chancery-lane, London, W.C. 2.—*Engineering.*

It is announced that the French Ministry of Public Instruction has decided to institute special courses of study with a view to training specialists in the working and use of Diesel engines and hot-bulb motors. Courses of instruction are to be given in the technical schools, and they will deal with both the practical and theoretical sides of the subject. As far as possible the needs of the various parts of the country will be considered, and special attention given to the application of such motors for marine purposes. A particular course of instruction is to be held in Paris for engineers and manufacturers who wish advanced information on the subject. According to a recent estimate there are over four million horsepower of Diesel engines in use, of which France possesses the modest share of some 150,000 hp., most of which is represented by land engines. The Government recognizes the apathy that exists with regard to Diesel engines, and considers the situation as one largely caused by the majority of the French nation being unaware of the qualities and possibilities of the heavy oil motor. It is hoped that the courses provided by the Ministry of Instruction will create a body of engineers and mechanics familiar with oil-engine practice and ready to further the use of such plant in motor ships and in industry generally.—*The Engineer.*

News in the Field of Power

City of Louisville Files Petition in Review

The city of Louisville, Ky., has requested the Federal Power Commission to set aside its ruling under which a preliminary permit was authorized for the Louisville Hydro-Electric Co. to develop power resources of the government dam at the Falls of the Ohio. The petition is a lengthy legal presentation, attempting to prove that the action of the commission is in error.

Industrial Accident Fatalities Decreasing

Immediate steps must be taken to prevent the ever-increasing and appalling toll of human lives taken by avoidable accidents in the United States, Marcus A. Dow, president of the National Safety Council, told the delegates to the Annual Safety Congress.

Accidents in 1922 claimed more than 75,000 lives in this country, Mr. Dow said. Over a period of ten years public-accident deaths have increased out of all proportion, while in the same period industrial accident fatalities have decreased.

The public safety section of the council estimated that during each day in 1922 there were 206 American citizens killed. Of this daily number 38 were automobile fatalities; 35 were caused by falls; 19 by drowning; 16 by burns; 9 by machinery accidents; 5 by street-car accidents and the remainder from miscellaneous causes.

Alabama Power Co. Denied Lock 17 Warrior River

Investigations by the staff of the Federal Power Commission have led to the conclusion that the commission would not be justified in granting a license to the Alabama Power Co. for its project at Lock 17 on the Warrior River. Reports prepared for the chief of engineers indicate that 80,000 primary horse power could be developed on the Warrior were three storage reservoirs constructed on upper tributaries. Assuming that the installation would amount to 200,000 hp., it is estimated that under those conditions water power could be made available for $80 per horsepower installed. Lock 17 is the key to the situation, as the construction of the reservoir would not be undertaken by anyone who does not have the right to develop Lock 17.

The commission engineers also were influenced by the apprehension that the development of the power resources of the Warrior might be delayed for an indefinite period owing to the fact that the Alabama Power Co. already has ahead of it large projects on the Coosa and Tallapoosa Rivers.

The company has been allowed until Dec. 1 to file any objections to the opinions expressed by the engineers.

Bureau of Standards Relief Fund Increasing

The permanent relief fund for the families of the employees of the Bureau of Standards who were injured or killed in the explosion in the altitude laboratory on Sept. 20, has received a check for $1,000 from Herbert Hoover, the Secretary of Commerce. The American Petroleum Institute, which was co-operating in the experimental work being done by these men at the time of the accident, has given a check for $10,000 for permanent relief; and other co-operating organizations are taking steps to give financial aid for the dependents of these men. The Accident Relief Committee of the Department of Commerce, empowered to receive and disburse funds for the purpose, consists of Dr. George K. Burgess, chairman; E. W. Libbey, secretary; Henry D. Hubbard, treasurer; and Dr. F. C. Brown.

Mill Wheel Site of 1634 Now Occupied by Modern Hydo Plant

In 1634, on the Lower Falls of the Neponset River, between Milton and Dorchester, Mass., then several miles from Boston but now a part of the suburb, Israel Stoughton built a small mill which he operated by water power and which appears to be the first recorded water-power development in America. As there are no detailed records, it is presumed that the mill was a grist mill and that the power was developed by means of an undershot or breast wheel constructed of timber, probably 15 to 20 ft. in diameter and possibly 10 to 15 ft. wide. The head of water is 8 to 9 ft., which would permit of a development of about 10 to 15 horsepower.

On the site of this old, original water-power development is now located one of Walter Baker & Co., Ltd., plants, which is one of the two new power-plants being installed, which are to be equipped with modern turbines developing 103 to 110 hp. with waterwheel governors for controlling the speed. They are direct-connected to generators of 75 kw. 600-volt to 100 kw. 2,300-volt capacity. The plant, which is being designed by D. F. Kilgore, of Boston, is built for complete automatic remote control and will be completed in about two months.

Coal Storage Is Balance Wheel Says Coal Commission

There have been various reactions to the Coal Commission's reports as well as strenuous objections to the findings by the Commission. Labor and operators object to the publicity of accounts and regulation; the public feels that it has been preyed upon by the coal industry and it would like to see somebody get punished. The non-punitive character of the report in no way appeases this thirst for revenge.

The coal consumers are told in the report on the "Storage of Coal" that storage by them is the all-important balance wheel between fluctuating consumption and variable production. Proper methods of storage of soft coal, the depth of piles, precautions against fires and spontaneous combustion, reclamation, loss of heating value, degradation are subjects on which the Bureau of Mines and state and private institutions can and will furnish technical information to those desiring it. The Federated Engineering Societies is preparing a report on coal storage addressed to the consuming public which the Coal Commission is co-operating in by making available to the engineers the detailed results of its investigation.

Girand License May Be Granted Oct. 15

When the Federal Power Commission meets on Oct. 15 it is expected that action will be taken on the Girand license covering the much-discussed project at Diamond Creek on the Colorado River. The prospects seem to favor the granting of the license. Secretary Wallace is openly in favor of such action. Under present conditions it might be expected that Secretary Work would vote against it. The deciding vote will be cast by Secretary Weeks. Chairman Weeks has been careful not to express any convictions in connection with the matter, but judging from questions he has asked, it would seem that he would favor the granting of the license provided it is made subject to the Colorado River compact or any other interstate agreement that may be made. There is also the possibility that conditions may be worked out between Mr. Girand and the representatives of the upstream states which would remove the objection of the latter.

Unquestionably, the strongest point being made in favor of the Girand li-

cense is the fact that the good faith of the federal government is involved. On the other hand, the upstream states argue that there is no legal requirement that the Government act precipitously, especially when the welfare of a great region is at stake. They think all questions of rights should be settled before any physical structures go in. In that connection it is claimed that the Arizona copper companies would suffer more by an unwise settlement of the Colorado Basin question than they will by the slightly longer period during which they will have to use higher cost power.

New Publications

Calculations in Heating and Ventilation. By G. S. Coleman, Director of the Department of Municipal and Sanitary Engineering in the Municipal College of Technology, Manchester, England. Published by Longmans, Green & Co., London and New York (Fourth Ave. at 30th St.) Cloth; 5½x8½ in.; 255 pages. Price, $5.

This book was originally prepared for the use of the author's advanced classes in heating and ventilation. The first fourteen chapters are given to a review of those fundamental principles of mechanics, hydraulics, heat, etc., that are necessary for a real understanding of the problems of heating and ventilation. The later chapters deal with heat transmission, direct and indirect radiation, humidity control, flow of air in ducts, fans and boilers. The aim throughout has been to lay the emphasis on the rational solution of the various problems involved rather than on descriptions of apparatus. Only elementary mathematics is involved.

Internal Combustion Engines, Theory and Design. By Robert L. Streeter. Published by McGraw-Hill Book Co., Inc., New York City, September, 1923. Cloth; 6 x 9 in.; 436 pages; 265 illustrations. Price $4.

For many years the study of gas-engine theory and design was under the severe handicap of lack of suitable textbooks in the English language. In 1915 Robert L. Streeter brought out the first edition of his treatise on Internal Combustion Engines, this work speedily winning recognition among the technical schools. Gas-engine design has undergone much change since that time, and in answer Mr. Streeter, since his present duties prevented personal attention to the subject, obtained the services of several teachers and engineers to revise the original work. One who is familiar with the first edition will not find a large amount of new matter. The Diesel engine has been given a little more space—probably all that is justifiable in a textbook on gas engines, but entirely too meager if the book is to be used on oil-engine design. Descriptions of a number of modern high-speed gas engines are included, as well as a chapter on gas producers. As a college textbook the second edition merits the praise given the first. The arrangement of material is logical, and the mathematics is not so involved as to prevent students familiar with calculus to follow the development. In addition to its use as a textbook there is so much useful information as to make the volume of value as a reference to those not having a deep grasp on mathematics.

Lectures on Engineering Practice 1922-1923 — Johns Hopkins University, School of Engineering. Published by Johns Hopkins Press, Baltimore, Md. Paper; 6x9 in.; 90 pages. Price, $1.

These lectures are reprints of lectures given before the Johns Hopkins University, School of Engineering, during 1922-23, and consist of the following: "The Relation of Highway Research to Modern Road Construction," by A. T. Goldbeck, Chief, Division of Tests and Research, United States Bureau of Public Roads, Washington, D. C.; "Possibilities for Steam Railroad Electrification," by Norman Wilson Storer, general engineer, Westinghouse Electric & Manufacturing Co., Pittsburgh, Pa.; "Power Development—Past and Future," by George A. Orrok, Consulting Engineer, New York City.

The Internal-Combustion Engine. By Harry R. Ricardo. Published by D. Van Nostrand Co., New York. Two volumes; cloth; 6½ x 10 in; 854 pages; 240 illustrations. Price, $18.

For many years the work of Clerk on the internal-combustion engine was the classic to which all those interested in the thermodynamic possibilities of the gas engine constantly referred. Much of the information given by Clerk has become obsolete by reason of later and more accurate experiments. Unfortunately, up to Ricardo's compilation of the two volumes on the internal-combustion engines, the results of these experiments had not been assembled in convenient form. Mr. Ricardo has done a praiseworthy piece of work, and while he speaks in generalities in most instances, there is much information that may be obtained from his treatise more conveniently than from research through old files of technical publications. The first volume deals with slow-speed engines, including both gas and oil, and also contains a concise exposition of the efficiencies obtainable from such engines. It is rather unfortunate that many of the illustrations are of engines no longer built. The sole Diesel shown as being built in the United States is that of the American Diesel Engine Co., which engine was discontinued in 1912. Likewise, his tests results cover trials of old units now supplanted by more modern designs. The second volume deals with high-speed gasoline engines and shows by its better treatment that the author is more experienced in this particular field. The two volumes will find their greatest field of usefulness as college text or reference books. For the designer there is a lack of definite facts as to engine proportions, etc. Those who are interested in internal-combustion engine developments will congratulate Mr. Ricardo on the results of what must have been a tremendous amount of labor.

Italian Government Finances. By H. C. MacLean. Trade Information Bulletin No. 130 Supplement to Commerce Reports, United States Dept. of Commerce, Government Printing Office, Washington, D. C. Free.

The Use of Statistics of Machinery Exports. By W. H. Rastall. Trade Information Bulletin No. 139, Supplement to Commerce Reports, United States Dept. of Commerce. Government Printing Office, Washington, D. C. Free.

Water-Power Projects

The Fox River Development — The State of Illinois will make an early application, it is understood, for a license covering development of water power on this river, having complied with the conditions of the preliminary permit.

The Interstate Public Service Co. has recently purchased land on the Ohio River near Jeffersonville, Ind., for the purpose of erecting, in the near future, an electric power plant which will furnish power and light for a large area of southern Indiana and which will also be tied in with big power plants which the Kentucky Utilities Co. has and is erecting in Kentucky. The plant at Jeffersonville is to have an ultimate capacity of 100,000 kw., but is at first only installing one 30,000 kw. turbine.

Green River, Utah, appears to be in line for early development of hydroelectric power. An application has been filed with the state engineer of Utah for a water permit for a combined power and irrigation project at the so-called Split Mountain site near Vernal. This application has been filed by a Mr. Ratliff, of Vernal, who also has requested the Department of the Interior to grant him an irrigation right to make the development under an outstanding filing made with the Land Office by him a number of years ago.

The Pigeon River Power Co. has applied for a license to construct a project on Big Pigeon River, in North Carolina, at the North Carolina-Tennessee state line. A diversion dam and tunnel will develop a head of 650 ft. and permit of an initial installment of 50,000 hp. Later developments will create storage in the river and its tributaries above the power site and will permit of an ultimate development of perhaps 150,000 hp. The power will supply the public-utility markets of Asheville and Knoxville and will bring industrial development to the power site. The Big Pigeon River is a tributary of the French Broad, which in turn is a tributary of the Tennessee, and because the ultimate project would affect navigation in the Tennessee and French Broad, the federal government claims jurisdiction.

Society Affairs

The Columbus, Ohio, Section of the A.S.M.E. will hear a paper by Harry E. Nold, professor of mine engineering, Ohio State University, "Coal Storage and the Management of the Coal Pile," on Oct. 26.

The **Providence Engineering Society** announces a joint meeting of the A.S.M.E. and Management Section on Oct. 23 at the society rooms at 8 p.m., a smoker for the evening of Nov. 2 and a meeting of the Power Section for Nov. 6, with a paper by Henry B. Oatley, of the Superheater Co., New York City, on "High-Pressure and High-Temperature Steam."

The **Society of Naval Architects and Marine Engineers**, 29 West 39th St., will have the following papers on the program of their thirty-first meeting which will interest readers of Power: "Fuel Conservation and Engineering Performance Standards by Shipping Board Vessels," by Joseph E. Sheedy; "Some Considerations on the Conversion of Steamships to Motorships," by Robert Haig.

The **Canadian Association of Stationary Engineers** through its committee on general education is formulating plans for a systematic educational program for the season of 1923-24. Combustion, heating, steam engineering, electricity and refrigeration are subjects that will be covered by lectures, and slides and general subjects such as political and social economy, business administration, etc., will be included.

Personal Mention

Dr. Samuel W. Stratton, president of the Massachusetts Institute of Technology, attended a meeting of the International Bureau of Weights and Measures in Paris during September.

Eugene Ogur, formerly with the Singer Manufacturing Co., who has recently been developing and promoting his automatic coal meter, is now with the United States Metal Refining Co., as test engineer at its refinery, Carteret, N. J.

L. B. Breedlove has been appointed consulting engineer to the bond department of the Chicago Trust Co. He is to analyze and to keep in intimate touch with the operation and management of public utilities and industrial properties, in the financing of which the bank is interested, with the view of rendering the maximum protection to investors.

Business Notes

Warren Webster & Co., Camden, N. J., manufacturer of heaters and heater meters, announces the opening of a new branch of the Chicago office at 708 West 34th St., Des Moines, Iowa. Hugo R. Sindlar, formerly of the Chicago, Ill., office, is to be in charge.

The **United American Metals Corp.**, Brooklyn, N. Y., announces that W. H. Jelley, formerly in the eastern Pennsylvania district, has been transferred to Pittsburgh and will be in charge of the Syracuse Babbitt Metals and Auto-crat Bushing Bronze sales for western Pennsylvania.

Black & Decker Manufacturing Co., Towson Heights, Baltimore, Md., manufacturer of air compressors, pumps, etc., announces a change in organization. Robert D. Black, will leave the Philadelphia office, being succeeded there by H. G. Smith, and become advertising manager, on Jan. 1, 1924, at Baltimore, Md.

Coming Conventions

The American Association for the Advancement of Science. Burton E. Livingston, Smithsonion Institution Bldg., Washington, D. C. Seventy-fifth anniversary meeting at University of Cincinnati, Cincinnati, Ohio, Dec. 27-Jan. 2.

American Electric Railway Engineering Association. James W. Welsh, 8 West 40th St., New York City. Annual Convention at Young's Pier, Atlantic City, Oct. 8-12.

The American Society of Civil Engineers. John H. Dunlap, 29 West 39th St., New York City. Fall meetings, Richmond, Va., Oct. 17-20.

American Society of Mechanical Engineers, Calvin W. Rice, 29 West 39th St., New York City. Annual meeting at New York City, Dec. 3-6. Regional meeting at Chattanooga, Tenn., Oct., 23-24.

American Society of Refrigerating Engineers, William H. Ross, 35 Warren St., New York City. Nineteenth Annual Convention at New York City, Dec. 3-5.

American Welding Society. M. M. Kelly, 33 West 39th St., New York City. Fall Meeting, Pittsburgh, Pa., Oct. 24-26.

Association of Railway Electrical Engineers. Jos. A. Andreucetti, Room 413, Chicago & Northwestern R.R. Station, Chicago, Ill. Thirteenth Annual Convention at Hotel La Salle, Chicago, Ill., Nov. 6-9.

Electric Power Club, S. N. Clarkson, 900 B. F. Keith Bldg., Cleveland. Fall meeting at French Lick Springs Hotel, French Lick, Ind. Nov. 19-22.

National Association of Practical Refrigerating Engineers; Ed. H. Fox, 5707 W. Lake St., Chicago. Fourteenth Annual Convention at Memphis, Dec. 12-16.

National Exposition of Power and Mechanical Engineering; Charles F. Roth, Room 1102, Grand Central Palace, New York City. Power Show, Dec. 3-8.

New England Association of Commercial Engineers; James F. Morgan, 53 Devonshire St., Boston, Mass. Power Show at Mechanics Bldg., Boston, Mass., Oct. 29-Nov. 3.

Society of Naval Architects and Marine Engineers, Daniel H. Cox, 29 West 39th St., New York City. Annual meeting at New York City, Nov. 8-9.

West Virginia-Kentucky Association of Mine, Mechanical and Electrical Engineers; Herbert Smith, Room 211, Robson-Prichard Bldg., Huntington, W. Va. Third annual convention at Frederick Hotel, Huntington, W. Va., Oct. 19-20.

Trade Catalogs

Stokers, Underfeed — Sanford Riley Stoker Co., Worcester, Mass. Catalog No. 82 contains 64 pages of description of stokers with illustrations, charts, cross-section drawings and tables.

Insulation — Mitchell-Rand Manufacturing Co., 18 Vesey St., New York City. Catalog No. 423, "Everything in Insulation," has the character of a handbook in that it contains a comprehensive statement about every material used in insulation from A to Z. Useful tables and a good index add decidedly to its value.

Engine-Gas — Foos Gas Engine Co., Springfield, Ohio. Bulletin No. 703 describes type T engine, which is of new design and intended for reserve or intermittent service for power-plant and general industrial work. Illustrations and condensed text cover the construction and operation of this engine, which is built in combinations of two, three, four, six and eight cylinders, 25 to 250 hp., 400 to 900 r.p.m.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal	Market Quoting	Sept. 24, 1923	Oct. 1, 1923
Pool 1,	New York	$3.00@3.50	$3.00@3.50
Smokeless,	Columbus	2.75@3.25	2.85@3.35
Clearfield,	Boston	1.85@2.50	1.85@2.60
Somerset,	Boston	2.00@2.75	2.00@2.75
Kanawha,	Columbus	1.75@2.10	1.75@2.00
Hocking,	Columbus	1.85@2.10	1.75@2.00
Pittsburgh No. 8	Cleveland	2.00@2.10	1.90@2.00
Franklin, Ill.	Chicago	2.75@3.25	2.75@3.00
Central, Ill.	Chicago	2.10@2.35	2.00@2.25
Ind. 4th Vein,	Chicago	2.50@2.75	2.50@2.75
West Ky.,	Louisville	1.75@2.10	1.75@2.00
S. E. Ky.,	Louisville	1.75@2.25	1.75@2.25
Big Seam,	Birmingham	1.75@2.15	1.75@2.15

FUEL OIL

New York—Oct. 4, light oil, tank car lots, 28@34 deg. Baumé, 3¾c. per gal., 36@40 deg. 4c. per gal., f.o.b. Bayonne, N. J.

Chicago — Sept. 28, 24@26 deg. Baumé, $1.82@2.07 per bbl.; 32@36 deg., $2.02@2.07 per bbl., tank cars.

St Louis—Sept. 25, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.60 per bbl.; 26@28 deg., $1.65; 28@30 deg., $1.75; 32@36 deg., gas oil, 3¼c. per gal.; 36@40 deg., distillate, 3¾c. per gal.

Pittsburgh—Sept. 27, f.o.b. local refinery, 30@34 deg., fuel oil, 5c. per gal.; 36@40 deg., fuel oil, 5½c.; 34 deg., neutral 7¼c.

Dallas—Sept. 28, f.o.b. local refinery, 26@30 deg., $1.35 per bbl.

Cincinnati—Sept. 25, tank-car lots f.o.b. local refinery, 26@30 deg. Baumé, 5c. per gal.; 30@32 deg., 5¼c.; 38@42 deg., distillate, 6½c.

Philadelphia—Sept. 27, 26@28 deg., $1.68@1.722 per bbl.; 16@20 deg., $1.64@1.68; 14@16 deg., $1.57@1.612.

New Plant Construction

PROPOSED WORKS

Ark., Fayetteville—The Southern Methodist Church, c/o Chautauqua Assn., is having preliminary plans prepared for the construction of various 2 and 4 story Chautauqua buildings, including hotel, auditorium, administration building, etc. Individual estimated cost from $25,000 to $150,000, total cost $500,000. Private plans.

Ark., Hiram—The Big Creek Power Co. is having plans prepared for the construction of an electric light plant, including a 40 ft. rock crib spillway dam; two 115 hp. turbines; two 75 kw. generators with exciters, transformers, arresters, switchboard, volt regulators, etc.; 3¾ mi. of transmission lines. Estimated cost $44,045. Private plans.

Calif., Byron—The Byron-Bethany Irrigation Dist., G. A. Howard, Secy., will receive bids until Oct. 15 for (1) furnishing and installing one horizontal centrifugal pump, 1 unit capacity, 9 sec. ft., 15 ft. head, 12 in. discharge, 575 r.p.m.; (2) steel discharge pipe; (3) two 14 in. gate valves; (4) one electric induction motor, 2,200 volt, 3 phase, 60 cycle, 40 hp., full load speed 575 r.p.m., with disconnect switch and starting compensator; (5) two complete electric priming units for plants 1 and 4; (6) concrete lining, etc.; (7) construction of irrigation ditches; (8) one wooden flood gate, Armstrong Creek; (9) concrete retaining walls near plants 1 and 4; (10) enlarging and reconstructing pump building, Sta. 4; (11 and 12) concrete pipe, also two outlet and one inlet gate with stop gate, etc.

Calif., Catalina Island (Los Angeles P. O.)—H. Gross, 936 South Alvarado St., representing a group of doctors, plans to build a hospital, etc., here. Cost will exceed $850,000.

Calif., Colusa—G. H. Farnsworth has had plans prepared for the construction of a pipe line; 11,000 ft. earth canal, 11 ft. top, 5 ft. wide on bottom, 2 ft. deep; two centrifugal pumps, capacity 13 sec. ft. and 7 sec. ft., 33 ft. maximum lift, operated by 50 and 100 hp. electric motor. Cost will exceed $45,000. F. H. Tibbetts, Alaska Commercial Bldg., San Francisco, Engr.

Calif., Cudahy (Florence P. O.) — The Zellerbach Paper Co., 220 South Los Angeles, plans to build a paper mill, here. Cost between $1,000,000 and $1,500,000.

Calif., Fairoaks—Beach & Cox plan to build 4 mi. canals, 4 ft. on bottom, 11 ft. wide on top, about 2 ft. deep, also to install two centrifugal pumps, 31.1 sec. ft. capacity, 12 ft. lift, operated by electric motors.

Calif., Glendale—The Glendale Hotel Co., c/o A. F. Priest, Archt., Fay Bldg., Los Angeles, is receiving bids for the construction of a 5 story apartment hotel and store building on California St. Estimated cost $250,000. Equipment detail not reported. Noted Aug. 14.

Cali., Los Angeles—The Guaranty Bldg. & Loan Assn., 6364 Hollywood Blvd., is having plans prepared for the construction of a 13 story, 75 x 150 ft. bank and office building on Hollywood Blvd. and Ivar St. J. C. Austin, Detwiler Bldg., Los Angeles, Archt. Equipment detail not reported.

Calif., Los Angeles—The Hellman Trust & Savings Bank, 601 South Main St., is having plans prepared for the construction of a 12 story, 57 x 150 ft. bank and office building on Hollywood Blvd. and Cherokee Ave. Estimated cost $80,000. Schultze & Weaver, Pacific Mutual Bldg., Los Angeles, Archts. Equipment detail not reported.

Calif., Los Angeles—The Insurance Exch. Bldg. Co. is having plans prepared for the construction of a 13 story 90 x 160 ft. office building on 9th and Olive Sts. Estimated cost $1,000,000. Cumett & Beelman, Union Bank Bldg., Archts. Equipment detail not reported.

Calif., Los Angeles—F. W. Johnson, et al., c/o Walker & Eisen, Archts., Pacific Finance Bldg., awarded the contract for the construction of a 14 story club rooms, loft and office building on Los Angeles and 17th Sts. to Edwards, Wildey & Dixon, Black Bldg., Los Angeles. Equipment detail not reported.

Calif., Los Angeles—W. W. Paden, et al., Story Bldg., is having preliminary plans prepared for the construction of a 12 story hotel on South Main St. Cost will exceed $750,000. L. L. Smith, Byrne Bldg., Los Angeles, Archt.

Calif., Los Angeles—The R. Wurlitzer Co., 605 West 7th St., is having plans prepared for the construction of a 12 story store and loft building on Broadway near 9th St. Cost will exceed $800,000. Walker & Eisen, Pacific Finance Bldg., Los Angeles, Archts. Equipment detail not reported.

Calif., Madera—The Central California Ice Co., Mono and P Sts., Fresno, plans to build an ice plant on a 50 x 100 ft. site on E St., here. Estimated cost $40,000.

Calif., Madera—A. J. Etter, et al., 409 North D St., plan to build an ice plant, 15 ton daily capacity, on 4th and E Sts. Estimated cost $40,000, including equipment.

Calif., Marysville—E. Glass, Archt., Underwood Bldg., San Francisco, will receive bids about Oct. 6 for the construction of a 5 story, 112 x 163 ft. hotel on 5th and E Sts., here, for the Marysville Hotel Co., c/o Architect. Estimated cost $300,000. Noted June 26.

Calif., Riverside — M. P. Waite, Story Bldg., Los Angeles, representing moving picture interests, plans to build a motion picture studio, here. Estimated cost $1,000,000.

Calif., San Francisco — The Bd. of Supervisors of the city and county of San Francisco voted for municipal distribution of the power developed on the Hetch Hetchy project and is having plans prepared for the construction of a transmission system to a point within the city limits and a complete distributing system. M. M. O'Shaughnessy, Engr.

Conn., Bridgeport—The D. M. Read Co., 1102 Main St., is having plans prepared for the construction of a 6 story department store on Broad and John Sts. Estimated cost $1,000,000. J. W. Northrop, 211 State St., Bridgeport, Archt. Monks & Johnson, 99 Chauncy St., Boston, Associate Archts. Equipment detail not reported. Noted Aug. 7.

D. C., Wash.—The Chief of Engineers, War Dept., will receive bids until Oct. 16 for 25 ventilating blowers complete with motors and starting regulators.

Fla., Jacksonville—The Gress Mfg. Co., Atlantic Natl. Bank Bldg., is in the market for a 75 hp. motor, 12,000 r.p.m.

Fla., Lake Worth—The city voted $40,000 bonds for a 500 kw. a.c., 3 phase, 60 cycle, 600 hp. addition to electric light plant. G. Krupley, Engr. Noted Sept. 4.

Ga., Dalton—The Amer. Thread Co. is having plans prepared for the construction of a 30,000 spindle thread mill, storehouse, boiler house, etc. on a 500 acre site. Lockwood, Greene & Co., Healy Bldg., Atlanta, Engrs. and Archts.

Ill., Chicago—The Flanders Apartment Hotel Corp., c/o A. S. Hecht, Archt., 155 North Clark St., is having plans prepared for the construction of a 146 x 275 ft. apartment hotel, including steam heating system, on Broadway and Buena St. Estimated cost $2,000,000.

Ill., Chicago—M. Holpuch, c/o Huszaah, Archt., 72 West Washington St., is having plans prepared for the construction of three 3 story, 330 x 600 ft. apartment houses, including steam heating system, on Drexel Ave. between 52nd and 53rd Sts. Estimated cost $2,000,000.

Ind., Crown Point—The city is having plans prepared for the construction of an electric light and power plant at the new city sanitarium. Estimated cost $50,000. K. D. Morris, Calumet Bldg., East Chicago, Ill., Engr.

Ind., Evansville—The North Star Coal Co., 301 Amer. Trust Bldg., is in the market for coal mining machinery, conveyors, etc.

Ind., West Lafayette (Lafayette P. O.)—Purdue University is having plans prepared for the construction of a 2 story, 120 x 150 ft. electrical laboratory building. Estimated cost $150,000, including equipment. Nicol, Scholer & Hoffman, 308 Main St., Lafayette, Archts.

Ky., Louisville—W. Heyburn, c/o Belknap Hardware & Mfg. Co., 2nd and Washington Sts., is having plans prepared for the construction of an 18 story, 132 x 200 ft. office building on 4th St. and Broadway. Estimated cost $2,000,000. Joseph & Joseph, Francis Bldg., Louisville, Archts. Equipment detail not reported.

Ky., Louisville—The Office of the United States Engineer, P. O. Box 72, is in the market for three turbo-generators, 500 kw., 110 volt.

Ky., Louisville—J. A. Wetmore, Acting Supervising Archt., Wash., D. C., will receive bids until Oct. 22 for the installation of one electric passenger elevator, hoistway, etc., in the United States Marine Hospital, here.

La., New Orleans—The Union Indemnity Co., 830-836 Union St., W. I. Moss, Pres., is having plans prepared for the construction of a 12 story office building on Baronne and Gravier Sts. Estimated cost $1,250,000. M. H. Goldstein, Hibernia Bank Bldg., and Favrot & Livaudais, Hibernia Bank Bldg., Archts. Equipment detail not reported.

La., Washington—The city is in the market for an electric light and power distribution system, including two 50 kw., 80% P. F., 60 cycle, 2,300 volt, a.c. generators direct connected to oil engines; switchboard; oil storage tank; vertical type service pump. Former bids rejected. Noted Sept. 4.

Mich., St. Joseph—The Whitcomb Hotel Co. is having plans prepared for the construction of a boiler house, etc. Cahill & Douglas, Engrs., 217 West Water St., Milwaukee, are receiving bids for a 300 hp. feed water heater and a vacuum pump.

Minn., Winona—The State Bd. of Control, Capitol, St. Paul, D. F. Mullen, Secy., received new bids for the construction of a 3 story, 200 x 210 ft. school building, including power plant, for the State Teachers' College, here, from I. N. Salyers, Estherville, Ia., $430,507; J. A. McDonald Constr. Co., 750 Builders' Exch., Minneapolis, $451,742; J. B. Nelson Constr. Co., Mankato, $455,000. Noted Sept. 25.

Mo., Cameron—The city voted $171,843 bonds, $53,433 for waterworks and distribution system; $53,860 for sanitary sewers; $64,550 for electric light plant and lines. Burns & McDonnell, 401 Interstate Bldg., Kansas City, Engrs. Noted Sept. 11.

Mo., St. Louis — The Grand-Franklin Realty Co., Grand and Franklin Aves., is having preliminary plans prepared for the construction of a 10 story, 103 x 170 ft. store and office building on Grand Blvd. Estimated cost $750,000. C. W. Martin, Syndicate Trust Bldg., St. Louis, Archt.

N. Y., Albany—The Albany City Hospital, North Scotland Ave., is having plans prepared and will receive bids about Oct. 17 for the installation of a steam heating plant and minor repairs to hospital for contagious diseases. Estimated cost $35,000. J. G. Brennan, City Hall, Engr.

N. Y., Canandaigua — The city, W. J. MacFarlane, Mayor, plans the installation of a lighting and pumping system. Cost between $60,000 and $75,000. Engineer not announced.

N. Y., Coney Island (Brooklyn P. O.)—The Seagate Hotel & Bath Co. will soon award the contract for the construction of a 13 story hotel on Surf Ave. and 24th St. Estimated cost $1,000,000. R. H. Segal, 110 West 40th St., New York, Archt. Noted Aug. 14.

N. Y., Hornell—The city is in the market for pumping machinery. An underground stream will be tapped and necessary machinery installed for pumping the water into the city mains. Estimated cost $10,000.

N. Y., New York—The Bd. Educ., 500 Park Ave., will soon receive bids for the construction of P. S. 116, including steam heating system, on Wren Pl. Estimated cost $1,000,000. W. H. Gompert, Flatbush Ave. Extension and Concord St., Brooklyn, Archt.

N. Y., South Amsterdam (Amsterdam P. O.)—The Adirondack Power & Light Co., Amsterdam, had plans prepared for the construction of an addition to its plant on the Mohawk River, here, and installation of a 40,000 hp. steam turbine generator unit. Estimated cost $3,000,000. Private plans.

N. C., Andrews—The Mayor and Bd. of Aldermen, G. Stillman, Clk., will receive bids until Oct. 24 for the construction of a hydro-electric plant on the Hiwassee River, including a concrete dam, 390 ft. long and 40 ft. high; power station; transmission line about 11 mi. long; furnishing hydraulic and electrical equipment, capacity about 1,250 kw. Ludlow Engrs., Winston-Salem, Engrs. Noted May 1.

Ohio, Avon Lake—The Cleveland Electric Illuminating Co., Illuminating Bldg., Cleveland, E. G. Crawford, Secy., is having plans prepared for the construction of an addition to its generating plant, here. Estimated cost $10,000,000. E. G. Cook, Illuminating Bldg., Cleveland, Engr. The owner is in the market for cables, wire and two generators with boilers to increase capacity by 75,000 hp.

Ohio, Cleveland — The city, c/o E. L. Myers, Dir. of Public Utilities, City Hall, is having plans prepared for the construction of additions to electric light plant on East 53rd St., also new equipment. Estimated cost $3,000,000. C. G. Beckwith, City Hall, Engr. Noted Oct. 31, 1922.

Ohio, Columbus—The Maple Grove Coal Co., 2269 Cleveland Ave., E. C. Riley, Mgr., is in the market for electrical equipment, including generator, etc., for new mine at Adena.

Ohio, Toledo—The Swan Creek Lumber & Supply Co., City Park Ave., K. Aschbacher, Purch. Agt., is in the market for unloading equipment, pit and automatic conveyors, etc.

Pa., Phila.—A. Kahn, Archt., 1000 Marquette Bldg., Detroit, is receiving bids and will open same about Oct. 10 for the construction of a 1 story, 400 x 920 ft. assembly plant, including steam heating equipment, on 63rd St., here, for the Ford Motor Co., Highland Park (Detroit P. O.).

Pa., Pittsburgh—The Pittsburgh Terminal R.R. & Coal Co., Wabash Bldg., is having plans prepared for the construction of outdoor substations and 22,000 volt transmission lines from here to Rook and Horning, W. Va. Crocelius & Phillips, Hanna Bldg., Cleveland, O., Consult. Engrs.

S. C., Columbia—The City Clerk, City Hall, will receive bids until Oct. 16 for the construction of a raw water pumping station. W. S. Tomlinson, City Hall, Engr.

S. C., Florence—The city, A. McTaggart, Clk., will receive bids until Oct. 10 for the installation of a heating plant in the city hall and opera house.

Tennessee—The Holston River Power Co., Johnson City, has filed application with the Federal Power Comn. for the construction of a dam, 90 ft. high, 25,000 hp. 6 mi. above Kingsport; a second dam, 35,000 hp. below confluence of the Wautauga and Holston Rivers; a third, 130 ft. high to produce 20,000 hp. above Bluff City.

Tenn., Memphis—H. S. Adler is having plans prepared for the construction of a 6 story, 160 room hotel on Linden Ave. near South Main St. Estimated cost $250,000. Hanker & Cairns, Scimitar Bldg., Archts. Equipment detail not reported.

Tenn., Memphis—The Bd. of Managers Methodist Hospital, J. R. Pepper, Chn., is having plans prepared for the construction of first unit of hospital, capacity 125 beds, on Union and Eastmoreland Sts. Estimated cost $300,000. W. F. Schulz, Chamber Commerce Bldg., Memphis, Engr.

Tenn., Memphis—E. Weil, Archt., Whitney Annex Bldg., New Orleans, La., and Hanker & Cairns, Associated Archts., Scimitar Bldg., Memphis, will receive bids until Oct. 15 for the construction of a 9 story, 150 x 150 ft. department store on Monroe Ave. and Main St., here, for the Lowenstein Department Store, controlled by the Maison Blanche Co., New Orleans. Separate bids will be received for elevators, heating equipment, etc.

Tenn., Nashville — Vanderbilt University, J. H. Kirkland, Chancellor, received lowest bid for the construction of a 5 story hospital and power plant on the campus from the Hegeman-Harris Co., 185 Madison Ave., New York. Estimated cost $2,000,000. Noted July 3.

Tex., El Paso—The Tri-State Lumber Co. is in the market for ice making machinery.

Utah, Fort Douglas (Salt Lake City P. O.)—The United States government plans to build fifteen training camp buildings. Constructing Quartermaster, Fort Mason, San Francisco, Archt.

Wash., Seattle—The Fraternal Order of Eagles, 7th and Pine Sts., is having plans prepared for the construction of a 3 story, 120 x 175 ft. lodge building, including clubrooms, offices, gymnasium and auditorium. Estimated cost $1,000,000. H. Bittman, Securities Bldg., Seattle, Archt. Equipment detail not reported.

W. Va., Maggie—The State Bd. of Control, Charleston, will receive bids until Nov. 30 for the construction of a 3 story, 45 x 201 ft. hospital with four 14 x 30 ft. projecting bays, including heating and ventilating plant, here. P. A. Davis, 3rd and Dunlap Sts., Phila., Pa., Archt.

Wis., Fond du Lac—The Bd. of Public Wks., J. F. Hohensee, Secy., will receive bids until Oct. 10 for the construction of an addition to sewage pumping plant, sewage ejector and additional machinery. G. H. Stanchfield, City Hall, Engr.

Wis., Janesville—Rock County, c/o M. L. Paulson, Evansville, Chn. Special Com., is having preliminary plans prepared for the construction of a 50 x 50 ft. power house, including boilers, feed water heater, chimney, etc., at the Poor Farm, here. Estimated cost $25,000. R. Kaupert, 290 3rd St., Milwaukee, Engr.

Wis., Watertown—A. Kuenzi, Engr., is receiving bids for a steam heating plant, piping, radiation and boilers for E. Miller. Estimated cost $25,000.

Wis., Wauzeka — The Kickappoo River Power Co., c/o E. E. Dillon, Engr., Washington Bldg., Madison, is having plans prepared for the construction of a hydro-electric power plant and dam to develop 1,700 hp., also transmission line and substations. Estimated cost $50,000.

Ont., Blenheim—S. B. Nichol, John St., is in the market for electric motors for planing mill.

Ont., Blyth—The Council, A. Milne, plans to install a hydro-electric lighting and power distribution system. The work will be carried out under the direction of the Ontario Hydro Comn., London, A. Beck, Chn. Estimated cost $28,000. The owner is in the market for complete equipment.

Every one of these items is reported by our authorized correspondents who are instructed to verify every item sent in. This free weekly service is published in the interests of the buyer and the seller, to bring them together and get machinery moving.

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BUSINESS NEWS DEPARTMENT
Tenth Ave. at 36th St., New York

Ont., Cayuga—The Council, A. Colter, Reeve, plans the installation of a hydro-electric system. Estimated cost $30,000.

Ont., London—The City Council, G. A. Wenige, Mayor, had plans prepared and will receive bids about Oct. 15 for the construction of a 2 story, 100 x 300 ft. city hall, including steam heating system, on Dundas St. Estimated cost $600,000. W. S. Pavey, Standard Bank Bldg., Archt. Noted May 30, 1922.

Ont., Lucknow—Silverwoods, Ltd., London, A. E. Silverwood, Genl. Mgr., plans to install a cold storage plant and equipment for the manufacture of ice cream at its branch plant, here. Estimated cost $35,000.

Alaska, Anchorage—The United States government has had plans prepared and will soon receive bids for the construction of a military post, including administration building, guard house, barracks, recreation hall, hospital, power houses, water, lighting and sewer systems, etc. Constructing Quartermaster, Fort Mason, San Francisco, Archt.

CONTRACTS AWARDED

Ark., Osceola—The Brown & Billings Co. will build by day labor, a 2 story ice plant, 40 ton daily capacity. Estimated cost $50,000.

Calif., Oakland—The Mutual Creamery Co., 3725 East 14th St., awarded the contract for the construction of a 1 story milk and ice cream plant on 5th Ave. and East 11th St. to M. E. Hooper, 1117 Webster St., Oakland. Estimated cost $43,000.

Ky., Louisville—The Office of the United States Engineer, P. O. Box 72, awarded the contract for steel and iron work for power house at Dam 44 to the Genl. Iron Wks. Co., Front and Harriet Sts., Cincinnati, Ohio. Estimated cost $5,000.

Mass., Holyoke—The Judd Paper Co., Race St., awarded the contract for the construction of a 3 story, 90 x 100 ft. paper storage, sales building, boiler room, garage, etc., to the Casper Range Constr. Co., 20 Bond St., Holyoke; heating contract to the Holyoke Valve & Hydrant Co. Estimated cost $125,000.

Minn., Westbrook—The village awarded the contract for a deep well unit, consisting of pump head, flanges, rods, well pipe, brass cylinder, etc., 50 g.p.m., to the Crane Co. of Minnesota, 5th St. and Broadway, St. Paul. Estimated cost $850. Noted Jan. 9.

N. J., Kearney—The Western Electric Co., 195 Broadway, New York, will build a power and manufacturing plant for telephone cables, here, by separate contracts under supervision of owner's engineers, McKenzie, Voorhies & Gmelin, 342 Madison Ave., New York, Archts.

N. Y., New York—The Bd. Educ., 500 Park Ave., awarded the contract for the construction of James Monroe High School on Boynton Ave. to the Willetts Constr. Co., 280 Madison Ave., (A) $2,057,000. Equipment detail not reported. Noted Oct. 2.

N. Y., New York—A. Campagna, 200 West 72nd St., will build and supervise separate contracts for the construction of a 15 story apartment house on Riverside Dr. and 89th St. Estimated cost $5,000,000. R. Candela, 200 West 72nd St., Archt. Equipment detail not reported.

N. C., Duke—The Erwin Cotton Mills awarded the contract for the construction of a 2 story mill, 35,000 spindle capacity, to J. L. Crouse, Greensboro. Estimated cost $500,000. Plans are also being prepared for a 1 story weave shed, 1,152 loom capacity, and other improvements at Mill 5. Total estimated cost $2,500,000. J. E. Sirrine & Co., South Main St., Greenville, S. C., Engrs. Mill 2 at present mechanically driven, will be electrified.

Okla., Cyril—The city, M. M. Rowe, Clk., awarded the contract for waterworks system, including well, tank, tower, hydrants, deep well pump and motor to the N. S. Sherman Co., 18 East Main St., Oklahoma City, $30,906.20. Noted Sept. 25.

S. C., Arcadia—The Arcadia Mills is building a 3 story, 105 x 280 ft. mill building, 50 x 78 ft. picker building, three 50 x 100 ft. warehouses, boiler house, switchboard room, pump house, etc. under supervision of Lockwood, Greene & Co., Piedmont Bldg., Charlotte, N. C., Engrs. and Archts.

S. D., Lemmon—The city awarded the contract for the construction of a standpipe to the Pittsburgh-Des Moines Steel Co., 9th and Tuttle Sts., Des Moines, Ia., $5,900; deep well to J. I. Case, Lemmon, $900. Other contracts not awarded. Noted July 3.

Wis., Appleton—The Water Comn., c/o Superintendent, City Hall, awarded the contract for two generators for pumping station to the Langstadt-Meyer Co., 735-737 Washington St., Appleton. Noted Aug. 7.

Wis., Biron (Grand Rapids P. O.)—The Consolidated Water Power Paper Co., Wisconsin Rapids, will build a 50 x 70 ft. boiler house and breeching by day labor, here. Estimated cost $40,000.

Wis., Madison—The University of Wisconsin, M. E. McCaffey, Secy., awarded the contract for the construction of steam tunnels from central heating station to the new hospital to G. Nelson, 1117 West Johnson St., Madison. Noted Aug. 28.

Wis., Milwaukee—The Lemm Electric Co., 3324 Fond du Lac Ave., awarded the contract for the construction of a 2 story, 34 x 80 ft. battery service station. Estimated cost $40,000. The owner is in the market for battery testing and charging equipment.

Wis., Neenah—The Lakeview Paper Co. awarded the contract for remodeling boiler house to C. R. Meyer & Sons, 50 State St., Oshkosh. Estimated cost $30,000. Orbison & Orbison, Engrs., College Ave., Appleton, will purchase new Wickes boilers and mechanical stokers.

Wis., Stratford—The Stratford Canning Co., c/o F. J. Mielke, Ripon, will build a cannery, consisting of a 48 x 106 ft. main building, 60 x 128 ft. warehouse and a 34 x 40 ft. power house, here, by day labor. Estimated cost $60,000.

Ont., Fonthill—The Council, A. M. Clark, Clk., awarded the contract for standpipe to the Horton Steel Wks., Bridgeburg, $7,500, pumps to the Canadian Fairbanks Morse Co., 26 Front St., Toronto. The remainder of the work, including construction of reservoir, pump house and water mains will be done by day labor. The owner is in the market for materials. Estimated cost $35,000. Noted Sept. 18.

Ont., Walkerton—The Council awarded the contract for standpipe to the Hunter Bridge & Boiler Co., Kincardine; gasoline engine and motor driven pumps to the Goldie McCulloch Co., Galt. Estimated cost $30,000. Noted Sept. 4.

PRICES - MATERIALS - SUPPLIES

Electrical prices on following page are prices to the power plant by jobbers in the larger buying centers east of the Mississippi. Elsewhere the prices will be modified by increased freight charges and by local conditions.

SINCE LAST MONTH

Declines—Raw linseed oil, $1.06 per gal. (5 bbl. lots) in Cleveland, against $1.18 last month; trend downward throughout country. Steel and tinned rivets and asbestos pipe and boiler covering discounts advanced considerably in New York warehouses. Chrome cement and magnesite brick (soaps and spits) down $1 per ton at plants. Prices lower on N. E. C. fuse cut-outs.

Advances—Consequent to recent advances in certain non-ferrous metals, antimony particularly, quotation on babbitt metal have advanced ½c. to 4½c. per lb. in Cleveland and 1c. to 2c. per lb. in Chicago warehouses. Armored cable, two cond., No. 14 solid, now $46 per M. ft. net, against $43, one month ago.

POWER-PLANT SUPPLIES

HOSE—

	Fire	50-Ft. Lengths
Underwriters' 2½-in., coupled		51½c. per ft.
Common, 2½-in., 3-ply		$1.00 per ft. list less 40–10–5%

	Air	
¼-in., per ft.	3-ply $0.33	4 ply $0.25

	Steam—Discounts from List	
First grade....40–5%	Second grade....45%	Third grade....50%

RUBBER BELTING—The following discounts from list apply to transmission rubber and duck belting:

Competition	60–5%	Best grade	50–10–5%

LEATHER BELTING—List price, 2c. per sq. in., per ply.

Grade	Discount from list
Medium	30–10%
Heavy	20–5–2½%

RAWHIDE LACING { For cut, best grade, 40%, 2nd grade, 50%. / For laces in sides, best, 48c. per sq.ft.; 2nd, 43c. / Semi-tanned: cut, 40%; sides, 48c. per sq.ft.

PACKING—Prices per pound:

Rubber and duck for low-pressure steam, ⅛ in.	$0.90
Asbestos for high-pressure steam, ⅛ in.	1.70
Duck and rubber for piston packing	.90
Flax, regular	1.10
Flax, waterproofed	1.70
Compressed asbestos sheet	.80
Wire insertion asbestos sheet	1.30
Rubber sheet	.45
Rubber sheet, wire insertion	.70
Rubber sheet, duck insertion	.50
Rubber sheet, cloth insertion	.25
Asbestos packing, twisted or braided and graphited, for valve stems and stuffing boxes	1.30
Asbestos wick, ½- and 1-lb. balls	.50

PIPE AND BOILER COVERING—Discounts, New York warehouses, are as follows:

85% magnesia high pressure		50%
For low-pressure heating and return lines	4-ply	74%
	3-ply	76%
	2-ply	78%

PORTLAND CEMENT—New York, $2.70@ $2.80 per bbl. without bags, in carload lots delivered on job. Bag charge of 40c. per bbl.

STRUCTURAL STEEL—New York delivered price, 3 to 15-in. beams and channels and 3 to 6-in. angles, tees, and plates, all $3.64 per 100 lb.

COTTON WASTE—The following prices are in cents per pound:

	New York Current	Cleveland	Chicago
White	10.00@13.00	15.00	11.00
Colored	8.00@13.00	12.00	8.00

WIPING CLOTHS—Jobbers' prices, in cents per lb., as follows:

	13½ x 13½	13½ x 20½
Chicago	10.00	13.00
New York	11.00	.....
Cleveland	$36.00 per M.	$52.00 per M.

LINSEED OIL—These prices are per gallon:

	New York	Cleveland	Chicago
Raw in barrels (5 bbl. lots)	$0.95	$1.06	$1.14

WHITE AND RED LEAD—In 100-lb. kegs, base price in cents per pound:

	Dry		In Oil	
	Current	1 Yr. Ago	Current	1 Yr. Ago
Red	14.00	12.75	15.50	14.25
White	14.00	12.75	14.00	12.75

RIVETS—The following quotations are allowed for fair-sized orders from warehouse:

	New York	Cleveland	Chicago
Steel ½ and smaller	50–10%	50–10%	60%
Tinned	50–10%	50–10%	4½c. per lb. net

Structural rivets, ¾, ⅞, 1 in. diameter by 2 in. to 5 in. sell as follows per 100 lb.

New York	$4.40	Chicago	$3.75	Pittsburgh	$3.00@3.25

Boiler rivets, same sizes:

New York	$4.50	Chicago	$3.85	Pittsburgh	$3.10@3.35

REFRACTORIES—Prices in carlots f.o.b. plant:

Chrome brick, eastern shipping points	net ton	$50@55
Chrome cement, 40@ 50% Cr_2O_3, in bulk	net ton	26@31
Chrome cement, 40@ 50% Cr_2O_3, in sacks	net ton	30@35
Magnesite brick: 9-in. straights	net ton	65@68
Magnesite brick: 9-in. arches, wedges and keys	net ton	71.50@74.80
Magnesite brick: Soaps and spits	per ton	91.00@95.20
Silica brick: Mt. Union, Pa.	per M	45.00@46.50
Clay brick, 1st quality, 9 in. shapes, Pennsylvania	per M	45@52
Clay brick, 1st quality, 9 in. shapes, Ohio	per M	45@46
Clay brick, 1st quality, 9 in. shapes, Kentucky	per M	45@46
Clay brick, 2nd quality, 9 in. shapes, Pennsylvania	per M	40@47
Clay brick, 2nd quality, 9 in. shapes, Ohio	per M	40@42
Clay brick, 2nd quality, 9 in. shapes, Kentucky	per M	40@42
Chrome ore crude, 40@50%	net ton	20.00@25.50

BABBITT METAL—Warehouse prices in cents per pound:

	New York	Cleveland	Chicago
(83% tin)	52.00	53.50	40.00
(35% tin)	25.00	17.50	17.00

COLD FINISHED STEEL—Warehouse prices are as follows:

	New York	Cleveland	Chicago
Round shafting and screw stock, per 100 lb. base	$4.65	$3.90	$4.55
Flats, square and hexagons, per 100 lb. base	5.15	4.40	5.05

BOILER SPECIALTIES—F. o. b. New York or Jersey City, discounts on list:

	Current
Copper ferrules	70%
Boiler flanges	60-10%
Boiler stay bolts	60%
Boiler patch bolts	10%
Boiler fitting-up bolts	45%
Pressed steel boiler lugs	10%
Pressed steel boiler hangers	10%

WROUGHT PIPE—The following discounts are to jobbers for carload lots on the latest Pittsburgh basing card:

BUTT WELD

Inches	Steel Black	Steel Galv.	Inches	Iron Black	Iron Galv.
1 to 3	62	50½	⅜ to 1½	30	13

LAP WELD

2	55	43½	2	23	7
2½ to 6	59	47½	2½ to 4	26	11
7 and 8	56	43½	4½ to 6	28	13
9 and 10	54	41½	7 to 12	26	11
11 and 12	53	40½			

BUTT WELD, EXTRA STRONG, PLAIN ENDS

1 to 1½	60	49½	⅜ to 1½	30	14
2 and 3	61	50½			

LAP WELD, EXTRA STRONG, PLAIN ENDS

2	53	42½	2	23	9
2½ to 4	57	46½	2½ to 4	29	15
4½ to 6	56	45½	4½ to 6	3	14
7 and 8	52	39½	7 and 8	1	7
9 and 10	45	32½	9 to 12	16	2
11 and 12	44	31½			

BOILER TUBES—Following are prices in New York warehouse of tubes manufactured according to specifications of the American Society of Mechanical Engineers:

Size	Lapweld Steel	C. C. Iron	Seamless Steel
4	.....	.....	$0.23
1¼	.....	.....	.25
1½	.....	.....	.22
1¾	$0.26	$0.29	.23
1	.20	.25	.21
1¼	.23	.29	.23
1½	.24	.33	.28
2¼	.....	.....	.....
2	.30	.41	.32
2¼	.32	.47	.35
3½	.34	.51	.37
2	.43	.64	.47

Tubes 2½ in. diameter, or smaller, over 18 ft. long, 10 per cent extra.
These prices are net per lineal foot based on stock lengths. If cut to special engths, billing will be based on the entire stock lengths.
In addition to the above, standard cutting charges are as follows:
1¼ in. to 2 in. diameter, 5c. per cut. 2½ in. diameter, 7c. per cut
2¾ in. diameter, 6c. per cut. 3 in. diameter, 9c. per cut
3¼ in. to 4 in. diameter, 10c.

ELECTRICAL SUPPLIES

ARMORED CABLE—Price per 1,000 ft.—5 per cent 10 days.

B. & S. Size	Two Cond. M Ft.	Three Cond. M Ft.	Two Cond. Lead M Ft.	Three Cond. Lead M Ft.
No. 14 solid	$ 46.00 (net)	$ 81.00 (net)	$164.00	$210.00
No. 12 solid	135.00	170.00	225.00	265.00
No. 10 solid	185.00	235.00	275.00	325.00
No. 8 stranded	285.00	375.00	520.00	500.00
No. 6 stranded	400.00	500.00	560.00	.....

From the above lists discounts are:

		Lead Covered
Less than coil lots	30%	15%
Coils to 1,000 ft.	40%	20%
1,000 to 5,000 ft.	45%	25%
5,000 ft. and over	45–5%	25–5%

CONDUIT, Price per 1,000 ft.; **ELBOWS AND COUPLINGS**, Per 100 pieces, f.o.b. New York, with 10-day discount of 5 per cent.

Size In.	Conduit Black Per M	Conduit Galvanized Per M	Elbows Black Per C	Elbows Galvanized Per C	Couplings Black Per C	Couplings Galvanized Per C
¾	$61.10	$66.16	$9.90	$11.03	$5.74	$6.16
¾	79.20	84.95	13.00	14.49	8.18	8.78
1	112.06	122.19	19.26	21.46	10.65	11.42
1¼	151.62	165.31	25.65	28.33	14.76	15.78
1½	181.28	197.67	34.19	37.77	18.23	19.47
2	248.90	265.94	62.70	69.24	24.29	25.97
2½	385.50	420.48	102.58	11.33	34.71	37.10
3	504.29	548.81	273.40	302.12	51.66	55.63
3½	633.86	688.65	604.03	667.15	69.41	74.19
4	772.62	838.59	697.06	770.99	86.76	92.73

CONDUIT BODIES AND FITTINGS—Black or galvanized.

	Less than $10 list	$10 list to $100	$100 list and over
Standard package	10%	20%	28%
Less than standard package	5%	10%	20%

CUT-OUTS—Following are net prices each in standard-package quantities:

CUT-OUTS, PLUG

S. P. M. L.	$0.10	D. P. D. B.	$0.32
D. P. M. L.	.16	T. P. S. B.	.30
T. P. M. L.	.24	T. P. D. B.	.42
D. P. S. B.	.18		

CUT-OUTS, N. E. C. FUSE

	0-30 Amp.	31-60 Amp.	60-100 Amp.
D. P. M. L.	$0.27	$0.75	$1.75
T. P. M. L.	.40	1.00	2.30
D. P. S. B.	.35	.90	.....
T. P. S. B.	.65	1.50	.....
D. P. D. B.	.65	1.75	.....
T. P. D. B.	1.15	3.00	.....
T. P. to D. P. D. B.	.80	2.10	.....

FLEXIBLE CORD—Price per 1,000 ft. in coils of 250 ft.:

No. 18 cotton reinforced heavy	$23.00
No. 16 cotton reinforced heavy	29.00
No. 18 cotton reinforced light	19.50
No. 16 cotton reinforced light	22.50
No. 18 cotton Canvasite cord	17.00
No. 16 cotton Canvasite cord	19.25

NATIONAL ELECTRIC CODE FUSES, NON-REFILLABLE—

250-Volt	Std. Pkg.	List	600-Volt	Std. Pkg.	List
3-amp. to 30-amp.,	100	$0.20	3-amp. to 30-amp.,	100	$0.40
35-amp. to 60-amp.,	100	.30	35-amp. to 60-amp.,	100	.60
61-amp. to 100-amp.,	50	.90	65-amp. to 100-amp.,	50	1.50
101-amp. to 200-amp.,	25	2.00	110-amp. to 200-amp.,	25	2.50
201-amp. to 400-amp.,	25	3.60	225-amp. to 400-amp.,	25	5.50
401-amp. to 600-amp.,	10	5.50	450-amp. to 600-amp.,	10	8.00

Discount: Less 1-5th standard package, 55%; 1-5th to standard package, 60%; standard package, 65%.

RENEWABLE FUSES, ENCLOSED—List price each:

Sizes	250-Volt List-Price	600-Volt List-Price	Std. Pkg. Quantity	Carton Quantity
1 to 30-amp.	$0.50	$1.10	100	10
35 to 60-amp.	1.00	1.25	100	10
65 to 100-amp.	2.00	3.00	50	5
110 to 200-amp.	4.00	5.00	25	5
225 to 400-amp.	7.50	11.00	25	1
450 to 600-amp.	11.00	16.00	10	1
450 to 600-amp.	11.00	16.00	10	1

REFILLS—

1 to 30	$0.30 ea.	$0.05	100	100
35 to 60	.05 ea.	.06	100	100
65 to 100	.10 ea.	.10	50	50
110 to 200	.15 ea.	.15	25	50
225 to 400	.30 ea.	.30	25	25
450 to 600	.60 ea.	.60	10	10

Discount Without Contract—Fuses:	
Broken carton	5%
Unbroken carton but less than std. pkg.	22%
Std. pkg.	40%
Discount Without Contract—Renewals:	
Less than std. pkg.	Net list
Standard package	40%
Discount With Contract—Fuses:	
Broken cartons	10%
Unbroken cartons but less than standard package	26%
Standard package	42%
Discount With Contract—Renewals:	
Less standard package	Net list
Standard package	42%

FUSE PLUGS, MICA CAP—

0–30 ampere, standard package (500)	$2.70
0–30 ampere, less than standard package	3.00

LAMPS—Below are present quotations in less than standard package quantities:

Straight-Side Bulbs Mazda B— Watts	Plain	Frosted	No. in Package	Pear-Shaped Bulbs or Bowl Enameled Mazda C— Watts	Clear	Frosted	No. in Package
10	$0.32	$0.37	120	75	$0.55	$0.60	60
15	.32	.37	120	100	.70	.75	24
25	.32	.37	120	150	.90	.95	24
40	.32	.37	120	200	1.15	1.25	24
50	.32	.37	120	300	1.75	1.85	24
60	.37	.42	120				
				500	2.50	2.65	12
				750	3.75	4.00	8
				1.000	4.25	4.50	8

Standard quantities are subject to discount of 10% from list. Annual contracts ranging from $150 to $300,000 net allow a discount of 17 to 40% from list.

PLUGS, ATTACHMENT—

	Each
Porcelain separable attachment plug	$0.19
Composition 2-piece attachment plug	.29
Swivel attachment plug	.12

RUBBER-COVERED COPPER WIRE—Per 1000 ft. f. o. b. New York.

No.	Solid Single Braid	Solid Double Braid	Stranded, Double Braid	Duplex
14	$ 7.00	$ 9.50	$11.22	$18.00
12	10.00	12.10	13.20	23.40
10	13.20	15.90	17.90	30.20
8	18.00	21.60	23.80	40.17
6	.....	.....	33.58	.....
4	.....	.....	45.00	.....
2	.....	.....	66.25	.....
1	.....	.....	88.50	.....
0	.....	.....	112.50	.....
00	.....	.....	137.00	.....
000	.....	.....	166.00	.....
0000	.....	.....	195.00	.....

SOCKETS, BRASS SHELL—

½ In. or Pendant Cap Key Each	Keyless Each	Pull Each	¾ In. Cap Key Each	Keyless Each	Pull Each
$0.33	$0.30	$0.55	$0.39	$0.36	$0.61

Less 1-5th standard package	25%
1-5th to standard package	32½%
Standard package	37½%

WIRING SUPPLIES—

Friction tape, ¾ in., less 100 lb. 34c. lb., 100 lb. lots	33c lb.
Rubber tape, ¾ in., less 100 lb. 34c. lb., 100 lb. lots	33c lb.
Wire solder, less 100 lb. 27c. lb., 100 lb. lots	29c. lb.
Soldering paste, 2 oz. cans	$1.00 doz.

ENCLOSED SWITCHES, KNIFE—Safety type, externally operated, 250 d.c. or a.c., N E.C.

TYPE "C" FUSED BOTTOM

Size, Amp.	Double Pole, Each	Three Pole, Each	Four Pole, Each
30	$4.50	$6.00	$7.25
60	7.50	8.25	10.50
100	10.50	13.00	22.50
200	16.00	20.00	36.00

Discounts:	
Less than $25.00 list value	30%
$25 to $50 list value	30–5%
$50 list value or over	35%